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ASSESSMENT OF THE APPLICABILITY OF AUTOMOTIVE EMISSION CONTROL TECHNOLOGY TO STATIONARY ENGINES

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ASSESSMENT OF THE APPLICABILITY OF AUTOMOTIVE EMISSION CONTROL TECHNOLOGY TO STATIONARY ENGINES

by

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The report gives a review of the emission characteristics of uncontrolled stationary diesel, spark ignition, and gas turbine engines, including an analysis and evaluation of the applicability of automotive emission control technology to stationary engines. Nitrogen oxides have been identified to be the principal pollutant species emitted from these engines. In principle, the emission control techniques developed or evaluated for spark ignition, diesel, and gas turbine engines are applicable to stationary engines. However, in most cases, the emission reductions achieved with these techniques are accompanied by sizeable losses in specific fuel consumption and uncertainties relative to the effect of these techniques on engine life and control system durability.

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ABSTRACT

This report gives a review of the emission characteristics of uncontrolled stationary diesel, spark ignition, and gas turbine engines, including an analysis and evaluation of the applicability of automotive emission control technology to stationary engines. Nitrogen oxides have been identified to be the principal pollutant species emitted from these engines. In principle, the emission control techniques developed or evaluated for spark ignition, diesel, and gas turbine engines are applicable to stationary engines. However, in most cases, the emission reductions achieved with these techniques are accompanied by sizeable losses in specific fuel consumption and uncertainties relative to the effect of these techniques on engine life and control system durability.

SECTION 1

HIGHLIGHTS

An examination and summarization was made of available information pertaining to (1) the exhaust emissions from uncontrolled stationary diesel, spark ignition, and gas turbine engines and (2) the automotive emission control techniques that have been developed to date and are potentially applicable to stationary engines. These techniques include engine derating; fuel injection and ignition timing retard; exhaust gas recirculation; catalytic converters; water injection; and engine component and operating condition modifications. A considerable amount of technical data relating to exhaust emissions, fuel consumption characteristics, and cost and effectiveness of various control techniques was obtained in the data acquisition process and is presented in the main body of the report. An analysis and evaluation of these data resulted in the following findings:

1. The oxides of nitrogen (NO_x) are the principal pollutant species emitted from stationary diesel, spark ignition, and gas turbine engines. Other pollutants emitted from these engines in various quantities include hydrocarbons (HC), carbon monoxide (CO), smoke, particulates, oxides of sulfur (SO_x), aldehydes, odor, and noise.

2. The HC and CO emissions in well-maintained diesel, gas turbine, and gas-fueled spark ignition engines are much lower than those from automotive spark-ignition engines. However, stationary gasoline engines operating at fuel-rich mixtures have HC and CO emissions that are comparable to automotive engines.

3. Smoke, particulate, odor, and SO_x emissions, while of some concern in diesels and gas turbines, are generally very low in spark ignition engines. SO_x is directly related to the sulfur content in the fuel and can be controlled by removing the sulfur during the refinery process.

4. In principle, automotive emission control techniques developed or evaluated for spark ignition, diesel, and gas turbine engines are applicable to their stationary counterparts. However, the degree of emission control realized and the resultant effect on fuel consumption, engine life, and operating parameters in stationary applications cannot be directly inferred from similar effects in automotive installations. Each such control technique must be evaluated with regard to the specific stationary engine type, design, and operating conditions.

5. Derating of stationary engines as a means of NO_x control, although effective in some cases, is not considered to be economically attractive because of the attendant increase in the cost of the engine per unit horsepower output and the potential degradation in specific fuel consumption. For example, derating a turbocharged divided chamber diesel engine by 30 percent to obtain a 40 percent reduction in NO_x would increase the investment cost per unit horsepower of the engine by 30 percent. This method may also result in increases in HC and CO emissions.

6. Fuel injection timing or ignition timing retard represent the only potential NO_x abatement techniques for diesel and spark ignition engines that require no hardware changes or additions. For this reason, these techniques are being considered by many manufacturers for application in these engines until better methods can be developed. From a cost-effectiveness point of view, injection/ignition retard is the least desirable approach, resulting in substantially higher specific fuel consumption and exhaust gas temperatures, (e.g., up to 12 percent increase in fuel consumption at 12 degrees injection retard for 50 percent reduction in NO_x). A limited amount of injection retard, combined with other techniques such as intake air cooling, might be feasible for

diesel engines to achieve a moderate NO_x reduction (approximately 25 percent) with essentially no loss in fuel economy.

7. Both exhaust gas recirculation (EGR) and water injection into the engine intake are very effective NO_x-abatement techniques for diesel engines, permitting NO_x reductions of up to about 60 percent. The small increase in specific fuel consumption associated with EGR can be compensated for, at least to some degree, by incorporating intake air cooling. In the case of water injection, the fuel consumption losses are generally negligible.
8. Water injection into the diesel engine cylinder, in the form of fuel/water emulsions, appears to be an attractive approach that merits further consideration primarily because of the greater simplicity of this technique relative to water induction. However, a number of potential problem areas related to corrosion and wear of important engine components would have to be resolved before incorporation of these techniques into stationary diesel engines.
9. Turbocharging of naturally aspirated diesel engines, combined with retarded injection timing and intercooling, results in NO_x reductions of up to 35 percent without any loss in specific fuel economy relative to the baseline configuration. The NO_x reduction actually achieved in various engines is largely dependent upon the degree of intercooling employed.
10. Variations in the fuel properties have very little effect on the NO_x emissions from diesel engines. In general, NO_x and HC decrease slightly with increasing cetane number while CO remains essentially constant. However, the fuel bound nitrogen contained in the heavier fuels might seriously limit the effectiveness of many potential NO_x emission control techniques.
11. Catalytic converters and thermal reactors are considered to have very limited applicability as HC and CO abatement devices in diesel engines because of the already low HC and CO concentrations and the contaminants contained in the diesel exhaust which might adversely affect catalyst durability. Thus, the use of oxidation catalysts would not be anticipated unless extremely low levels of HC and CO emissions were required. More significantly, neither of these devices is effective in reducing NO_x because of the excess air contained in diesel exhaust.

12. Achievement of diesel engine NO_x emissions substantially lower than those obtainable through incorporation of the emission control techniques discussed above requires the development of new engines. These would incorporate certain modifications in the combustion chamber geometry, injectors, timing schedules, turbocharger, and intercooler. The development of such an engine is a difficult task that would require several years.

13. Although no Federal emission standards are currently in existence for stationary gas turbines, several states are enforcing the Federal standards that have been promulgated for steam powerplants, and a number of counties and cities have formulated even more stringent regulations. With few exceptions, the current gas turbines are capable of meeting these limits by incorporation of modest combustor modifications or addition of water or steam injection. However, these techniques are limited in their effectiveness and are insufficient to achieve substantially lower emission standards.

14. The modest combustor modifications required to meet current load regulations have a negligible effect on the investment and operating cost of gas turbines, particularly in the case of large units. On the other hand, water or steam injection can more than double the investment cost of small units (below 1 MW output) and increase the cost of large units (30 MW or above) by seven to ten percent.

15. Future emission control efforts in gas turbines are expected to concentrate on the development of low-emission "dry" combustors utilizing the advanced low NO_x combustion technology which is currently being developed for potential use in automotive gas turbines. The ultimate goal of these developments is a premixed, prevaporized, well-stirred combustor. It is estimated that the development time of such combustors would be between five and ten years.

16. Externally mounted gas turbine combustors may offer more flexibility and greater potential for emission control than conventional in-line combustors, particularly for heavy fuels and residuals.

17. Catalytic combustors, having a potential of low emissions, are of interest to automotive gas turbines but are not considered applicable to most stationary gas turbines at this time because of many unresolved potential problem areas

such as: limited catalyst durability, uncertain specific heat release rate, ignition characteristics, and high pressure drop.

18. In principle, the emission control techniques developed to date for automotive spark-ignition engines are applicable to stationary gasoline and gas-fueled spark ignition engines. However, in most cases the emission reductions achieved with these techniques are accompanied by sizable losses in specific fuel consumption and substantially increased operating cost. For example, in one case utilization of water injection to reduce NO_x 60 percent resulted in a six-percent increase in specific fuel consumption.

19. On the basis of current state-of-the-art technology, the most cost effective NO_x emission control technique projected for use in stationary gasoline and gas spark ignition engines appears to be a combined system consisting of optimum valve and port timing, intake charge cooling, intake air humidification, and operating speed changes. However, a slight increase in the emission of hydrocarbon might occur in this case which then might further require the use of a thermal reactor or catalytic converter.

20. It is difficult to evaluate the relative need for implementing any of the various control techniques shown to be applicable to stationary engines. This difficulty arises principally because there is no accurate inventory of emissions for the various engine classes and no perspective as to how these emissions relate to emissions from other sources and air quality. In addition, no goals have been set for stationary engine emission reduction (aside from local regulations for gas turbines) and the future trends of utilization of the various stationary engine types is uncertain.

SECTION 2

INTRODUCTION

This report presents a compilation, evaluation, and assessment of available information pertaining to the applicability of automotive emission-control technology to stationary diesel, spark ignition, and gas turbine engines, both in retrofitting current engines and application to new engines.

To fulfill the objectives of this study, the work effort was divided into two basic phases: the first phase was concerned with the compilation and review of applicable information acquired from: (1) an open literature survey and (2) visits to engine manufacturers and users for technical discussions of relevant engine and emission control system performance characteristics and economic factors. In the second phase of the study, a summarization and evaluation of all data acquired in the first phase was made.

The results of this study are presented in the following order and context: Section 3 is concerned with stationary engine characteristics and addresses the various engine design approaches, applications, emissions, and specific fuel consumption. Diesel engines are discussed in Subsection 3.1, spark ignition engines in Subsection 3.2, and gas turbines in Subsection 3.3.

Section 4 discusses the effectiveness of all known emission-control techniques/devices considered by the automotive industry in their efforts to meet current and future emission control standards. Whenever possible, data from stationary engines are included and compared with corresponding automotive engine data, both in terms of emission reduction and fuel consumption effects. Subsection 4.1 is devoted to diesel engines, Subsection 4.2 to spark ignition engines, and Subsection 4.3 to gas turbines.

Section 5 presents an evaluation of the emission control approaches identified in Section 4 with respect to performance and economics.

Section 6 identifies those areas where further research and development (R&D) efforts are needed to bridge existing data gaps and provide the technical information required for a more comprehensive assessment of the cost effectiveness of various emission control approaches.

Appendix A presents a summary table listing the installed horsepower of all stationary diesel, spark ignition and gas turbine engines. Appendix B lists those organizations which contributed to this study, either directly by providing useful engine test data, or indirectly through general discussions of engine performance and emission characteristics. Appendix C presents metric system conversion factors.

SECTION 3

STATIONARY ENGINE CHARACTERISTICS

In this section of the report, brief discussions are presented relative to the design and application of stationary diesel, spark ignition, and gas turbine engines. The major portion of the section is concerned with the emissions aspects of these three engine types, both from a rated-load and part-load point of view. Although the oxides of nitrogen have been identified to be the principal pollutant species from these engines, other emissions including hydrocarbons, carbon monoxide, oxides of sulfur, aldehydes, smoke, particulates, odor, and noise, were given limited consideration.

Since there is a lack of emission data from stationary engines, particularly with respect to large size/low speed designs, applicable emission data from heavy-duty automotive engines are included in the emission characterization of several of the engine categories considered in this section. Because of many similarities in the design and operating parameters of heavy-duty automotive and large stationary engines, this approach appears to be justified.

3.1 DIESEL ENGINES

3.1.1 Engine Description

The diesel or compression ignition engine is a reciprocating engine in which air is compressed in the cylinder and the fuel is then injected into the hot air toward the end of the compression stroke. Numerous 4- and 2-stroke engine configurations have been designed around the basic thermodynamic diesel cycle, and a number of these are currently being marketed by many manufacturers throughout the world. The various designs can be grouped into two categories: open-chamber (or direct-injection engines) and divided-chamber (or indirect-injection engines). Each of these categories can then be subdivided into a number of engine classes, depending upon the type of combustion chamber and air induction system utilized.

Many diesel engines are fitted with turbochargers to increase their power output and to improve the specific fuel consumption. Frequently, an aftercooler is added between the compressor exhaust and the inlet manifold to reduce the air charge temperature, thus raising the peak power output of the engine.

Because of their superior fuel consumption characteristics, direct-injection engines are favored by many manufacturers for use in stationary and heavy-duty truck applications. Conversely, the divided-chamber configurations are preferred by the makers of light-duty automotive diesels and by some manufacturers of stationary and heavy-duty truck engines.

Briefly, in open-chamber diesel engines, fuel is injected several degrees before the piston reaches top dead center. To enhance fuel-air mixing and improve combustion, a certain degree of air swirl is generated in the combustion chamber by means of specially designed intake manifolds, valves, and pistons. Depending upon the magnitude of the swirl in the combustion chamber, the direct-injection engines can be subdivided into the following three classes: (1) quiescent or low-swirl "Mexican hat" chambers, favored by many domestic manufacturers, (2) medium-swirl, deep-bowl chambers, used primarily in Europe, and (3) high-swirl, spherical "M" combustion chambers, used in special applications.

The divided-chamber diesel engine employs two combustion chambers, consisting of a main chamber and an antechamber connected to the main chamber through a communicating flow passage. This engine category can be divided into three classes: (1) swirl chamber or turbulence chamber--a configuration favored by a number of European and Japanese light-duty and heavy-duty diesel engine manufacturers, (2) precombustion chamber or prechamber, employed in a number of European and domestic light-duty and heavy-duty diesel engines, and (3) air cell and energy-cell combustion chamber.

The important design features of these six engine classes are briefly described in the following sections.

3.1.1.1 Open-Chamber, Low-Swirl Diesel

The low-swirl, open-chamber diesel engine utilizes a shallow dish (or a "Mexican hat") combustion chamber. In the absence of an induced air swirl, mixing of the fuel and the air is accomplished by means of a sophisticated fuel injection system. One or more injection nozzles are utilized, each having several fuel injection orifices (Refs. 3-1 through 3-4).

This engine class is favored for large low-speed applications (300-1200 rpm), but is also used in some medium-size, medium-speed (1200-2500 rpm) designs. Normally the engine is operated with considerable excess air; and a high combustion efficiency is generally achieved even with low-grade fuels.

Because of the low-air swirl in the chamber and the lean air-fuel mixture operation, the heat losses to the cylinder wall and head are minimized, resulting in low specific fuel consumption, desirable starting characteristics, and moderate cooling system requirements. Other features of this particular engine class include long exhaust valve life (low exhaust gas temperatures) and residual fuel capability.

3.1.1.2 Open-Chamber, Medium-Swirl Diesels

Open-chamber, medium-swirl configurations are utilized primarily in engines in the medium-power regime (less than 100 hp per cylinder) operating at speeds up to about 3000 rpm. In these engines, the amount of fuel injected during each power stroke is smaller than in the case of the previously discussed low-swirl engines. Therefore, fuel-air mixing by means of multiple fuel sprays is no longer possible, and incorporation of some air swirl is required to complete the mixing process within the short time span available. The air swirl is generated by one of several approaches, including the use of directed intake ports or masked intake valves (Ref. 3-5). Typically, air-swirl ratios (defined as the rotational speed of the air swirl in the chamber just prior to ignition, divided by the rotational speed of the engine) between 3 and 6 are employed in this engine class (Ref. 3-1).

In general, medium-swirl, open-chamber diesels exhibit good fuel economy combined with acceptable starting characteristics and low heat losses to the chamber wall.

3.1.1.3 Open Chamber, High Swirl Diesels

The high-swirl diesel, or "M" engine, utilizes air-swirl ratios of the order of 12 generated by means of a corkscrew-type intake port and the squish action of the special piston (Ref. 3-6). In general, this engine utilizes a single injector nozzle which supplies a coarsely atomized spray into the spherical combustion chamber. Most of the fuel is carried to the wall by centrifugal forces where it is deposited in the form of a thin film. The remainder of the fuel (approximately five percent), which has been finely atomized during the injection process, is then ignited. As combustion proceeds, the flame front progresses to the wall region of the chamber and ignites the vaporized fraction of the fuel initially deposited on the wall in liquid form. The burning rate of this fuel is controlled by the fuel vaporization process and, as a result, relatively low maximum combustion pressures and pressure rise rates are obtained. Even at engine speeds above 3000 rpm, the pressure rise rates in this design are comparable to medium-swirl diesels.

The principal advantage of this engine is its multifuel capability. Since combustion is vaporization-controlled, almost any type of fuel can be burned, and the engine runs very quietly even on gasoline. The major drawback of the engine is its poor cold-start capability which is related to the high heat transfer into the chamber wall during compression.

3.1.1.4 Divided-Chamber, Swirl-Chamber Diesel

The swirl chambers employed in these engines are spherical or semi-spherical in shape and have a capacity of the order of 50 percent of the total clearance volume (Ref. 3-7). During the compression stroke, air is forced from the main chamber into the swirl chamber through the narrow passage connecting the two chambers and a high degree of air swirl is then generated by the inrushing air. The fuel is injected into the swirl chamber and is deposited on the chamber walls by the centrifugal forces set up by the air swirl. Some of the larger fuel droplets follow the air swirl along the wall, while the smaller ones start to evaporate immediately. Ignition occurs in the vicinity of the chamber throat and diffusion-flame-type combustion spreads from there throughout the swirl chamber. As the pressure in the swirl chamber rises, the hot combustion gases are discharged into the main chamber where sufficient excess air is available to complete the combustion process. Upon leaving the swirl chamber, a strong secondary air swirl is generated which enhances flow mixing in the main chamber and assures high combustion efficiency.

Relative to open-chamber configurations, swirl-chamber engines have a number of inherent advantages. These include higher speed capability, lower fuel sensitivity, lower maximum pressures, lower pressure rise rates, lower temperatures and lower exhaust emissions.

The principal disadvantage of the swirl-chamber diesel is its higher fuel consumption, which is primarily due to the higher surface-to-volume ratio of the combustion chamber and the high swirl velocities. Furthermore, during a cold start, a sizable fraction of the total compression heat is transferred to the swirl-chamber wall by the high air swirl, and starting aids such as glow plugs are required to minimize engine cranking time.

3.1.1.5 Divided-Chamber, Prechamber-Diesel

Although there are significant design differences, the operating characteristics of the prechamber diesel engine are, in many respects, similar to the previously discussed swirl chamber.

The prechamber is generally pear-shaped, rather than spherical, and has a capacity of only 20 to 30 percent of the total clearance volume. The passage connecting the prechamber and the main chamber is more restricted than in the case of the swirl chamber. This increases air turbulence in the main chamber and tends to confine the initial shock occurring during combustion to the prechamber (Ref. 3-8).

In general terms, the advantages and disadvantages of prechamber diesel engines relative to open-chamber designs follow those of the swirl chamber. The specific fuel consumption of the prechamber engine is expected to be slightly lower than for the swirl chamber, because of the lower heat losses associated with the lower turbulence levels and physical size of the prechamber.

3.1.1.6 Divided-Chamber - Air-Cell Diesels

The air-cell diesel engine employs a small antechamber located in line with the fuel injection nozzle and connected to the main chamber by a small orifice-type restriction. Upon ignition in the main chamber, additional air is forced into the air cell. As the piston descends, the air trapped in the air cell is gradually readmitted into the main chamber generating considerable turbulence and improving the combustion efficiency. Since the advent of the swirl chamber, interest in the air cell concept has declined.

The energy cell or Lanova cell concept is a modified version of the simple air cell configuration and combines the features of the prechamber and the air cell. The principal feature of the energy-cell diesel is its smooth operation. The cold-start characteristics of the engine are favorable and the specific fuel consumption is acceptable, but higher than that of equivalent open-chamber configurations.

3.1.2 Design Considerations

Generally, stationary engines are designed for minimum total operating cost (initial cost, fuel consumption, and maintenance); long life; good response and overload capability; and low noise and vibration levels. The useful life of large, low-speed engines is of the order of 30 years, with major overhauls every 20,000 hours. Minor overhauls (inspection, injector cleaning, etc.) are scheduled at 10,000-hour intervals.

For medium and small stationary diesels, the time between major overhauls has been quoted by one manufacturer to be about 5000 to 8000 hours.

3.1.3 Applications

Diesel engines are widely used by electric and natural gas utilities, the petroleum industry, and many operators of small electric power and pumping stations.

Many electric utilities employ diesel engines as prime movers of continuous- and peaking-power generators, standby power installations and, more recently, total energy systems. Many of the transmission line and process compressors utilized by the petroleum industry are powered by diesel engines. Also, diesel engines are frequently used by the petroleum industry as drives for oil and gas well drilling and pumping equipment, water pumps, and electric generators. The operators of small electric power stations and pumping installations include municipalities and commercial firms which are utilizing diesel engines to supply part of their electric power needs and to power total energy systems and water and sewage pumping units.

Generally, the large (above 1000 hp), low-speed diesel engines marketed by a number of manufacturers are designed for continuous operation as frequently required in stationary applications. Conversely, with few exceptions, the medium (100 to 1000 hp) and small (below 100 hp) stationary diesel engines are modified versions of heavy duty on-highway and off-highway truck engines. To increase their service life these engines are then operated at derated power and speed settings. The modifications incorporated into the engines might include a new head, governor, and fuel injection system.

3.1.3.1 Installed Power

In 1971 the total estimated installed horsepower of stationary reciprocating engines was 34.7×10^6 bhp (excluding gasoline engines). Of these, liquid-fueled diesel engines contributed 11.8×10^6 bhp and gas diesels contributed about 4.1×10^6 bhp with the remainder supplied by spark ignition engines. A breakdown of the installed diesel engine horsepower for 1971 is presented in Table 3-1 (Ref. 3-9).

3.1.3.2 Operating Modes

A review of the available data on electric power capacity and power generation from all sources in 1970 indicates that reciprocating engines, operated on liquid and gaseous fuels, represent about 1.2 percent of the total electric generating capacity in the United States and about 0.3 percent of the total power generated (Ref. 3-9). The capacity factor of these engines as a whole is only about 12 percent, indicating that many engines are utilized only for short periods typical of electric peaking power and standby installations. In this context,

TABLE 3-1. ESTIMATED INSTALLED DIESEL ENGINE HORSEPOWER FOR 1971[a] (Ref. 3-9)

Application	Diesel Fuel hp	Dual Fuel hp	Total Installed Power, hp
Electric Power Generation[b]	1,570,000	3,710,000	5,280,000
Oil and Gas Pipelines	830,000	390,000	1,220,000
Oil and Gas Exploration	1,500,000	-	1,500,000
Agricultural	7,500,000	-	7,500,000
Water and Sewage	465,000	-	465,000
Total	11,865,000	4,100,000	15,965,000

[a] A summary table listing the installed horsepower of all stationary engines is presented in Appendix A.

[b] Estimated 1970 data.

the capacity factor is defined as the ratio of the total energy generated per year by an engine or engine class to their total energy generation capacity. The projected capacity factors for 1980 and 1990 are 8 percent and 7 percent, respectively (Ref. 3-9).

In continuous power installations, diesel engines are typically operated at 80 to 90 percent of their rated power for about 6000 to 8000 hours per year. In peaking installations, the engine operates normally for several hours per day near full load, whereas the standby units are generally run for 1 to 3 hours per week.

In 1971 the total energy generated by diesel engines used in oil and gas pipeline installations was 5000 x 10^6 bhp-hr for liquid fueled engines and 2,350 x 10^6 bhp-hr for diesels operated on natural gas. Based on the total installed power figures listed in Table 3-1, the capacity factor of these engines was about 69 percent which is in reasonable agreement with data provided by one engine manufacturer.

As shown in Table 3-1, the total diesel horsepower installed by the oil and gas exploration industry in 1971 was 1,500,000 bhp. These engines generated 1.94 x 10^6 bhp-hr, resulting in a capacity factor of about 15 percent (Ref. 3-9).

The capacity factors assumed by Shell (Ref. 3-9) for agricultural pumping and municipal water and sewage pumping are 40 and 75 percent, respectively. Frequently, the agricultural-pumping units are operated near their intermittent rating load levels for periods extending from 1 to 10 days.

3.1.3.3 Fuel Requirements

Liquid-fueled diesel engines are generally operated on No. 2 diesel fuel, although heavier fuels have been used with good success in some of the large low-speed engines. One manufacturer has stated that attempts to run his engines on heavy fuels proved to be unsuccessful because of excessive deposit buildup in the exhaust ports and fuel injectors. Other potential problem areas noted by several manufacturers are related to the high sulfur and metal content, normally found in heavy fuels, which could be detrimental to engine life. In general, fuels having a sulfur level of 0.5 percent or less are considered to be acceptable, whereas higher concentrations are not recommended because of the danger of excessive engine wear and buildup of harmful acid compounds in the lubricating oil.

Dual fuel diesel engines are frequently utilized by the electric power generation and petroleum industries. In these engines approximately 90 to 95 percent of the total heat input is supplied by natural gas and the remainder by distillate fuels such as No. 1 or No. 2 diesel fuel or fuel oil. The objective of the liquid fuel is to initiate ignition in the combustion chamber at the desired crank angle position of the piston.

The principal advantages of gas diesels relative to oil diesels include the lower cost of gaseous fuels and the longer life of the lubricating oil. However, in view of the increasing shortage of natural gas, it may not be possible to maintain the fuel cost differential much longer. In general, gaseous fuels result in lower pressure rise rates and peak pressures in the combustion chamber which is desirable from an engine durability point of view.

3.1.4 Emissions

Diesel engines, like other heat engines, emit a number of different pollutant species, including hydrocarbons, carbon monoxide, oxides of nitrogen, oxides of sulfur, aldehydes, particulates, smoke, and odor. Except for the oxides of sulfur and nitrogen these compounds are the result of imperfect or incomplete combustion. To provide a better understanding of the problems related to diesel engine emissions, the gaseous pollutants and their formation in diesels are briefly described in the following section.

3.1.4.1 Pollutant Formation

3.1.4.1.1 Hydrocarbons

The hydrocarbons (HC) emitted from diesel engines are considered to be the result of quenching of oxidation reactions occurring in the wall region of the combustion chamber and in ultra-lean zones of the air-fuel mixture (Refs. 3-2 and 3-10 through 3-12). Factors influencing the level of HC emissions include the degree of turbulence inside the combustion chamber, and certain design details in the injection and combustion system geometry affecting the formation of the fuel spray (Ref. 3-2). Relative to spark-ignition engines, the HC emissions from diesels at rated conditions are low but tend to increase with decreasing load. This is further discussed in Section 3.1.4.2.

3.1.4.1.2 Carbon Monoxide

Carbon monoxide is the result of a deficiency in oxygen during the combustion process. While the overall air-fuel mixture ratio in diesel engines is lean, oxygen deficient regions exist throughout the combustion chamber. Although sufficient oxygen is ultimately available to complete the CO reactions, this oxygen may not reach the CO molecules before the temperature of the gases in the chamber drops to a value too low for oxidation to proceed. In general, the CO emissions from diesel engines are rather low at full load and decrease further as engine load is reduced (increasing air-fuel ratio).

3.1.4.1.3 Oxides of Nitrogen

The oxides of nitrogen are the principal pollutant species emitted from diesel engines. Their formation during the combustion process is kinetically controlled and increases with increasing temperature, oxygen concentration, and residence time of the gases in the chamber (Ref. 3-2).

As further discussed in Section 3.1.4.2, the NO_x emissions from prechamber engines are generally lower than from existing open chamber diesels. In prechamber engines, combustion begins under fuel-rich conditions in the prechamber near the top dead center (TDC) position of the piston. Because of a lack of oxygen, very little NO is formed under these conditions. As combustion proceeds in the prechamber, the burning charge is expanded into the main chamber where an adequate amount of oxygen is available to complete the reactions. Since the air in the main chamber is relatively cool, the NO formation reactions are quenched rapidly, hence minimizing NO_x (Refs. 3-2 and 3-13). In addition, the rapid rate of energy release in divided-chamber engines provides for optional timing at a more retarded position compared to open-chamber engines.

In open-chamber engines, combustion starts several crank degrees before TDC. Since the injection delay period is relatively long in these engines, there is an appreciable amount of vaporized fuel available at the time of ignition. This results in very rapid combustion followed by compression heating leading to high local flame temperatures and NO_x formation rates (Refs. 3-13 and 3-14).

Insufficient information is currently available regarding the effects of fuel-bound nitrogen on the NO_x emissions from diesel engines. However, it is well known that in utility boilers up to 70 percent of the bound nitrogen is converted to NO_x (Ref. 3-15).

3.1.4.1.4 Oxides of Sulfur

The oxides of sulfur emissions from diesel engines are directly related to the sulfur content of the fuel. In the absence of adequately developed instrumentation, it is generally assumed that all sulfur oxidizes to sulfur dioxide. Thus, the mass emissions of SO_x are twice the mass of sulfur contained in the fuel (Ref. 3-16).

3.1.4.1.5 Aldehydes

Aldehyde emissions, which are the result of partial oxidation of hydrocarbons in the combustion process, may be a contributor to diesel engine odor. Test data indicate that many diesel engines emit significant amounts of aldehydes.

3.1.4.1.6 Particulates and Smoke

Particulate matter found in the exhaust of diesel engines consists of impurities contained in the fuel and unburned- or partially-burned hydrocarbon molecules. Frequently, carbon particles are emitted from diesel engines in the form of smoke which is formed in fuel-rich zones having sufficiently high temperature to decompose the fuel. In general, smoke levels of well adjusted diesels are low except, perhaps, at full load where the air-fuel ratio approaches stoichiometric. Smoke and CO are formed under much the same conditions and their concentrations are generally proportional because both are dependent upon the availability of oxygen (Ref. 3-2).

3.1.4.1.7 Odor

The odor from diesel engine exhaust has long been recognized as undesirable. Although the odor formation mechanism has not yet been fully explained, there is evidence that the diesel odor is the result of partial oxidation reactions taking place in the lean regions of the combustion chamber.

3.1.4.2 Test Procedures and Instrumentation

To simulate the operation of the engine in the field, the emission test procedures employed by most manufacturers of heavy-duty diesel engines and by independent test laboratories are based on measurements at steady-state operating conditions. Emission maps have been determined for many engines by operating the engine at a number of discrete speed and load settings. In particular, this procedure is utilized for engines used in both mobile and stationary installations. Conversely, the emission data available for large stationary engines are frequently limited to their rated speed/load points.

For some engines, the available emission data are based on the 13-mode procedure which has been developed to characterize the HC, CO, and NO_x emissions of heavy-duty engines used in on-highway truck applications (Ref. 3-17). In this procedure the engine is operated at 13 points (11 discrete sets of speed and load) and the specific mass emissions over the cycle are then determined by multiplying the mass emissions obtained at each mode point by an appropriate weighting factor and dividing the sum of these values by the average power output of the engine.

The instrumentation employed in diesel engine emission test work includes nondispersive infrared (NDIR) analyzers for the measurement of CO and NO concentrations and heated flame-ionization-detectors (FID) for HC, to prevent condensation of the heavier HC species in the sampling lines. More recently, the chemiluminescence analyzer has been added by some investigators to measure NO and NO_x separately. Tests conducted by the Southwest Research Institute on locomotive diesel engines indicate relatively small differences in the NDIR and chemiluminescence readings (Ref. 3-18).

Aldehydes are normally determined by wet chemistry and particulates are measured by gravimetric methods.

Full-flow light extinction type analyzers, such as the PHS smoke meter, are preferred for diesel exhaust smoke measurements because of their good accuracy and fast response. In cases where this technique would be difficult to implement, other instruments such as the Bosch and AVL smoke meters have been applied with good success (Refs. 3-2 and 3-19).

3.1.4.3 Gaseous Emissions

This section includes a discussion of the HC, CO, and NO_x emission characteristics of current uncontrolled heavy-duty diesel engines. The emissions at rated engine operating conditions are presented in Section 3.1.4.3.1 and the part load emissions are discussed in Section 3.1.4.3.2. Although many of the engines considered in the study were originally designed for mobile applications, their emission characteristics are considered to be applicable to stationary engines as well, because of many design and operational similarities between the various diesel engine classes.

The emission data and correlations presented in the following sections are based on data published in the open literature or were acquired directly from various engine manufacturers.

3.1.4.3.1 Rated Conditions

The emissions at rated conditions of many naturally aspirated and turbo-charged, open-chamber and divided-chamber, four- and two-stroke diesel engines are presented in Tables 3-2 through 3-7, in terms of pollutant specie concentrations and specific mass emissions. Smoke, odor, air-fuel ratio, and specific fuel consumption data are also listed in these tables and these parameters will be discussed in later sections of the report. For many engines, the data shown represent average values from two or more test runs.

The emissions from eleven four-stroke, naturally aspirated, open-chamber diesel engines tested by a number of investigators are listed in Table 3-2. Ten of these engines were originally designed for mobile (on- or off-highway) applications, but several of them are offered also for use in stationary installations. Engine No. 11

TABLE 3-2. EMISSIONS FROM FOUR-STROKE, NATURALLY ASPIRATED OPEN CHAMBER DIESEL ENGINES - RATED CONDITIONS

Manufacturer	Engine Identification No.	CID	No. of Cylinders	Power, bhp		Speed, rpm		Concentration, ppm			Specific Mass Emission, g/bhp-hr			Smoke Opacity, %	Odor Intensity, DI Units[a]	Specific Fuel Consumption, lb/bhp-hr	Air/Fuel Ratio	Reference
				Rated	Measured	Rated	Measured	HC	CO	NO_x	HC	CO	NO_x					
General Motors	1	478	6	155	152	3200	3200	1000	1400	740	2.2	6.1	5.3	16	4.5	0.395	24.1	3-12
Mack	2	678	6	185	172	2100	2100	630	1400	2550	1.3	5.6	16.6	2	6.4	0.421	26.2	3-12
Caterpillar	3	573	6	-	186	-	3000	420	1100	1300	0.8	4.2	8.3	5	5.9	0.357	24.5	3-12
-	4	-	-	-	172	-	2100	680	1000	2250	1.3	3.9	14.4	-	6.9	0.402	24.5	3-12
International Harvester	5	407	6	112	110	2400	2500	930	1900	1456	1.9	7.5	9.5	-	-	0.405	21.4	3-16
International Harvester	6	-	6	200	175	3000	3000	470	540	1030	1.0	2.2	6.8	7	-	0.412	21.2	3-20
Caterpillar	7	-	-	225	210	2800	2800	200	2075	1200	0.4	7.3	6.9	14	-	0.410	18.6	3-20
Cummins	8	-	-	260	242	2100	2100	71	1493	1231	0.1	5.3	7.2	19	-	0.433	17.8	3-20
Perkins	9	354	6	112	104	2800	2800	230	3300	1130	0.5	14.6	8.3	20	5.9	0.470	20.7	3-12
-	10	-	-	-	209	-	2100	210	2100	840	0.5	8.7	5.6	-	4.6	0.464	31.7	3-12
-	11	Large	-	-	-	low	-	-	-	-	0.5	4.2	15.2	-	-	-	-	-

[a] Diesel Intensity

TABLE 3-3. EMISSIONS FROM FOUR-STROKE, TURBOCHARGED, OPEN CHAMBER DIESEL ENGINES - RATED CONDITIONS

Manufacturer	Engine Identification No.	CID	No. of Cylinders	Power, bhp		Speed, rpm		Concentration, ppm			Specific Mass Emission, g/bhp-hr			Smoke Opacity, %	Odor Intensity, DI Units[a]	Specific Fuel Consumption, lb/bhp-hr	Air/Fuel Ratio	Reference
				Rated	Measured	Rated	Measured	HC	CO	NO_x	HC	CO	NO_x					
Mack	12	673	-	232	232	2100	2100	1140	1000	1440	2.9	5.2	12.2	2	4.0	-	22.4	3-12
Allis Chalmers	13	426	-	157	148	2200	2200	137	643	1510	0.3	2.7	10.6	-	-	0.406	23.0	3-16
John Deere	14	404	6	129	136	2200	2200	844	530	1158	2.1	2.6	9.3	-	-	0.410	25.8	3-16
Mack	15	673	-	235	228	2100	2100	423	720	1601	1.0	3.4	12.5	3.5	-	0.392	26.7	3-20
Mack	16	865	-	325	290	-	2400	331	518	1488	1.1	3.3	15.6	2.8	-	-	29.6	3-20
Cooper Bessemer	17	Large	12	4300	4300	600	600	-	-	-	0.1	3.9	11.0	-	-	-	-	3-21
Cooper Bessemer	17[b]	Large	12	4300	4300	600	600	-	-	-	5.2[c]	4.5	9.0	-	-	-	-	3-21
-	18	Large	-	-	-	Low	-	-	-	-	0.8	2.8	9.0	-	-	0.395	31.3	-
-	18[b]	Large	-	-	-	Low	-	-	-	-	0.5	2.4	15.6	-	-	6400[d]	34.0	-
-	19	Large	-	-	-	Low	-	-	-	-	0.5	1.1	13.1	-	-	-	-	-
-	19[b]	Large	-	-	-	Low	-	-	-	-	0.5	1.1	11.0	-	-	-	-	-

[a] Diesel Intensity
[b] Gas - diesel
[c] Not used in averaging
[d] Btu/bhp-hr

TABLE 3-4. EMISSIONS FROM FOUR-STROKE NATURALLY ASPIRATED, DIVIDED CHAMBER DIESEL ENGINES - RATED CONDITIONS

Manufacturer	Engine Identification No.	CID	No. of Cylinders	Power, bhp		Speed, rpm		Concentration, ppm			Specific Mass Emission, g/bhp-hr			Smoke Opacity, %	Odor Intensity, DI Units[a]	Specific Fuel Consumption, lb/bhp-hr	Air/Fuel Ratio	Reference
				Rated	Measured	Rated	Measured	HC	CO	NO_x	HC	CO	NO_x					
Daimler-Benz	20	108	4	29	28	2400	2400	123	2025	280	0.3	9.1	2.1	-	-	0.528		3-14
Daimler-Benz	21	121	4	60	57	4200	4200	55	832	601	0.1	3.3	4.0	-	-	0.526		3-22

[a] Diesel Intensity

TABLE 3-5. EMISSIONS FROM FOUR-STROKE, TURBOCHARGED, DIVIDED CHAMBER DIESEL ENGINES - RATED CONDITIONS

Manufacturer	Engine Identification No.	CID	No. of Cylinders	Power, bhp		Speed, rpm		Concentration, ppm			Specific Mass Emission, g/bhp-hr			Smoke Opacity, %	Odor Intensity, DI Units[a]	Specific Fuel Consumption, lb/bhp-hr	Air/Fuel Ratio	Reference
				Rated	Measured	Rated	Measured	HC	CO	NO_x	HC	CO	NO_x					
Caterpillar	22	-	-	285	279	2200	2200	80	200	700	0.2	0.9	5.0	2.0	3.6	0.425	24.7	3-12
Caterpillar	23	638	6	149	142	1900	1900	23	101	545	0.1	0.5	4.3	-	-	0.399	26.2	3-16
Caterpillar	24	-	-	245	246	2200	2200	36	52	664	0.1	0.3	5.0	2.0	-	0.399	25.1	3-20
Caterpillar	25	-	-	285	285	2200	2200	-	-	-	0.2	0.7	4.3	-	-	-	-	3-23

[a] Diesel Intensity

TABLE 3-6. EMISSIONS FROM TWO-STROKE, OPEN CHAMBER, BLOWER SCAVENGED DIESEL ENGINES - RATED CONDITIONS

Manufacturer	Engine Identification No.	CID	No. of Cylinders	Power, bhp		Speed, rpm		Concentration, ppm			Specific Mass Emission, g/bhp-hr			Smoke Opacity, %	Odor Intensity, DI Units[a]	Specific Fuel Consumption, lb/bhp-hr	Air/Fuel Ratio	Reference
				Rated	Measured	Rated	Measured	HC	CO	NO_x	HC	CO	NO_x					
General Motors	26	426	-	208	208	2100	2100	130	585	1675	0.4	3.6	17.1	-	-	0.422	33.0	**3-16**
General Motors	27	284	4	141	141	2100	2100	170	1000	1350	0.5	5.6	12.5	-	3.6	0.413	29.5	**3-12**
General Motors	28	-	-	-	-	-	2100	199	1134	1392	0.6	7.1	14.4	1.5	-	0.421	33.2	**3-20**

[a] Diesel Intensity

TABLE 3-7. EMISSIONS FROM LARGE TWO-STROKE, OPEN CHAMBER, TURBOCHARGED DIESEL ENGINES - RATED CONDITIONS

Manufacturer	Engine Identification No.	CID	No. of Cylinders	Power, bhp		Speed, rpm		Concentration, ppm			Specific Mass Emission, g/bhp-hr			Smoke Opacity, %	Odor Intensity, DI Units[a]	Specific Fuel Consumption, lb/bhp-hr	Air/Fuel Ratio	Reference
				Rated	Measured	Rated	Measured	HC	CO	NO_x	HC	CO	NO_x					
General Motors	29	10,320	16	3200	3208	900	900	236	567	945	0.6	3.0	8.0	-	-	0.392	-	**3-18**
-	30	Large	-	-	-	-	-	-	-	-	0.3	1.6	8.3	-	-	0.379	37.0	-
-	31	Large	-	-	-	-	-	-	-	-	0.3^b	1.6^b	9.0^b	-	-	6900^c	31.5	-

[a] Diesel Intensity
[b] Dual fuel
[c] Btu/bhp-hr

represents a large low-speed diesel engine which is utilized in stationary installations only. As indicated, the emission concentrations and specific mass emissions show considerable variations. On a percentage basis, the largest variations are observed for HC, reflecting the previously noted emission sensitivity to injector and combustion chamber design details. HC and CO of the large engine (No. 11) are considerably lower than the average emissions of this engine class, while NO_x is substantially higher. Attempts to correlate the emissions with different engine parameters such as speed, air-fuel ratio, and brake mean-effective-pressure (torque) proved to be unsuccessful, primarily because of a lack of sufficient data from large engines.

Table 3-3 presents the emissions at rated conditions for eight four-stroke, turbocharged, open-chamber diesels. Engines No. 12 through 16 are utilized primarily in mobile applications whereas Engines No. 17, 18, and 19 are large-size, low-speed stationary engines which are operated either on No. 2 diesel fuel or natural gas. The average NO_x specific mass emission of the stationary engines is about 12 g/bhp-hr compared to an average value of about 11.5 g/bhp-hr for the "mobile" engines. This difference is considerably smaller than the observed engine to engine variability in NO_x. The HC and CO emissions of the large diesels are somewhat lower than the average levels obtained for the smaller "mobile" units. With respect to fuel effects on emissions, the data indicate very little variations in HC, CO, and NO_x when operating on No. 2 diesel fuel or natural gas. Comparison with the data in Table 3-2 indicates similar average HC emissions for the naturally aspirated and turbocharged engines; whereas CO of the turbocharged engines is considerably lower, reflecting the higher air-fuel ratio used in this engine type. Conversely, the average NO_x of turbocharged diesels is higher than for naturally aspirated engines. This is attributed to the higher intake manifold temperature associated with turbocharging, which results in higher compression and peak combustion temperatures, hence higher NO_x formation rates.

Test data from two small four-stroke naturally aspirated divided-chamber diesel engines are listed in Table 3-4. As expected from theoretical considerations, the NO_x emissions of these engines are markedly lower than those of the open chamber engines shown in Tables 3-2 and 3-3, and HC is extremely low as well.

The emissions from four four-stroke, turbocharged, divided-chamber diesel engines manufactured by Caterpillar Tractor Company are listed in Table 3-5. Again the HC and CO emissions are extremely low, owing to the staged combustion process employed in divided-chamber diesels. The NO_x emissions of these engines are considerably lower than in open-chamber engines but somewhat higher than for the naturally aspirated divided-chamber engines listed in Table 3-4.

Two-stroke, open-chamber, blower-scavenged, and turbocharged diesel engine data are presented in Tables 3-6 and 3-7, respectively. The blower-scavenged engines were manufactured by General Motors and are used primarily in mobile applications, while the turbocharged engines represent large, low-speed stationary designs. Compared to the open-chamber, four-stroke engines, the medium-size blower-scavenged, two-stroke diesels have lower HC, comparable CO but higher NO_x emissions. Conversely, the average emissions of the large turbocharged two-stroke diesels tend to be somewhat lower than for the four-stroke designs. However, when making comparisons between the various diesel engine categories, the small data sample currently available for some of the engine categories must be taken into consideration.

3.1.4.3.2 Part Load Emissions

HC, CO, and NO_x exhaust emission concentrations of several four-stroke, naturally aspirated, open-chamber diesel engines are plotted in Figures 3-1 and 3-2 as a function of air-fuel ratio. Also shown in Figure 3-2 are brake-mean-effective-pressure (load) data correlated with air-fuel ratio. At full load the air-fuel ratio of four-stroke naturally aspirated diesels varies between about 19 and 24, but decreases rapidly with decreasing load. As shown in Figure 3-1 the HC emission concentrations show considerable engine to engine variability but seem to be essentially independent of air-fuel ratio. The CO exhaust concentrations have a maximum in the low air-fuel ratio regime corresponding to high engine loads and decrease rapidly, as the mixture is leaned out. However, increasing CO concentrations are observed in some engines under very lean, light-load operating conditions which is attributed to quenching effects. As expected, the NO_x concentrations reach a maximum at rated engine conditions and decrease rapidly with decreasing load (increasing air-fuel ratio), as a result of a reduction in the quantity of air involved in the combustion process. Similar trends were obtained for other diesel engine classes.

Typical part load HC, CO, and NO_x emission curves for the various diesel engine designs are presented in Figures 3-3 through 3-9 in terms of specific mass emissions versus percent of maximum engine power. Unlike the data listed in Tables 3-2 through 3-7, the curves shown in these figures represent individual test runs.

The specific mass emission characteristics of a typical four-stroke naturally aspirated, open-chamber diesel engine are illustrated in Figure 3-3. Design and operational details of this engine are listed in Table 3-2. As indicated, HC increases steadily with decreasing load and speed. This trend follows from the constant HC concentrations shown in Figure 3-2 and the higher exhaust flow rates per

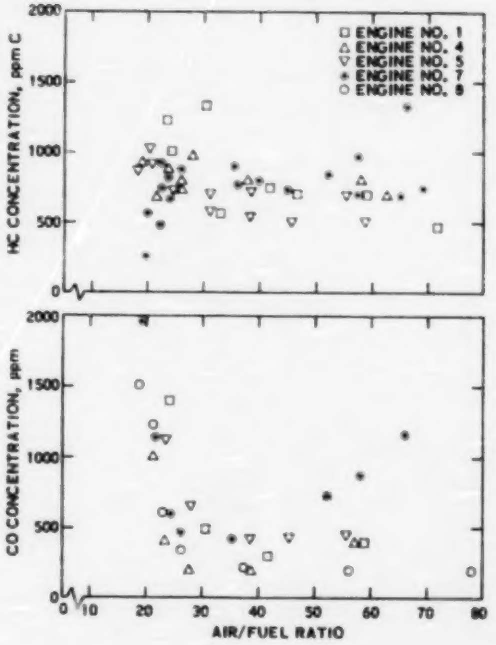


Figure 3-1. HC and CO emissions vs air-fuel ratio - four-stroke, naturally aspirated, open-chamber diesels

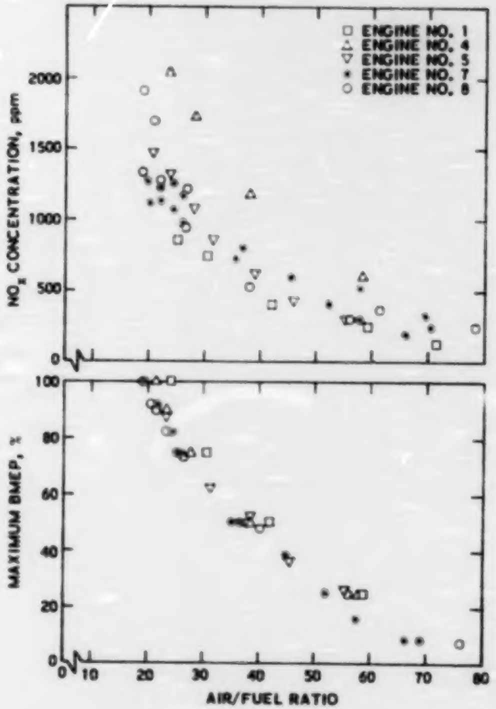


Figure 3-2. NO_x emissions and brake mean effective pressure vs air fuel ratio - four-stroke, naturally aspirated, open-chamber diesel engines

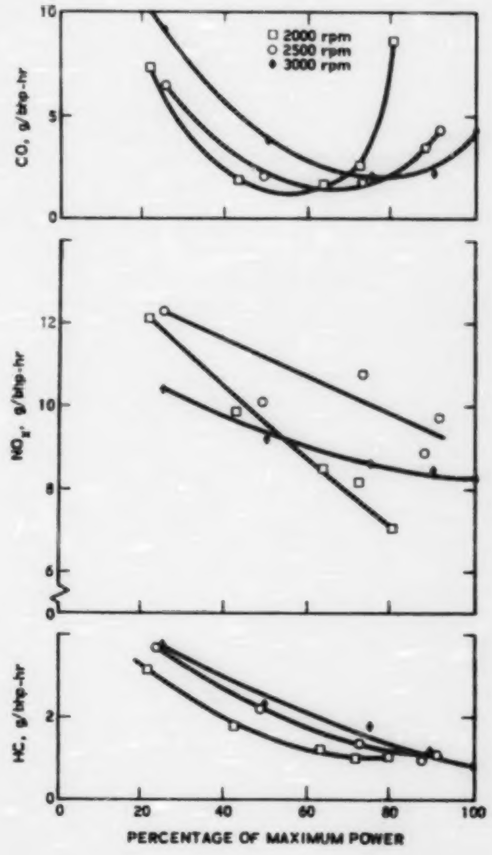


Figure 3-3. Specific mass emissions - four-stroke, naturally aspirated, open-chamber diesel (Engine No. 3)

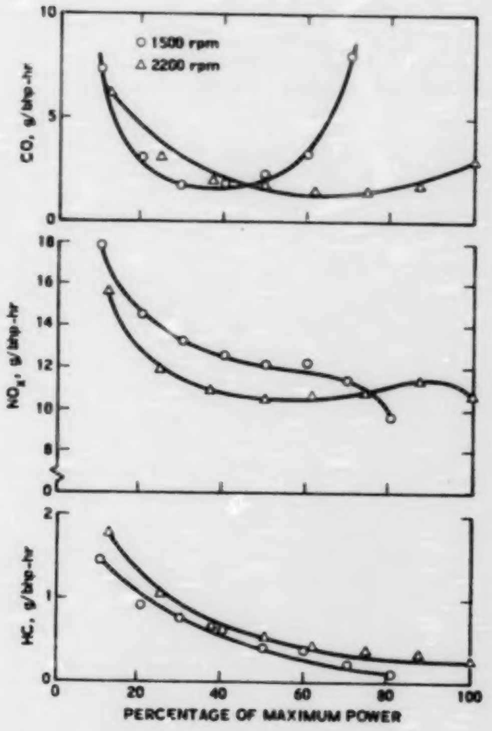


Figure 3-4. Specific mass emissions - four-stroke turbocharged, open-chamber diesel (Engine No. 13)

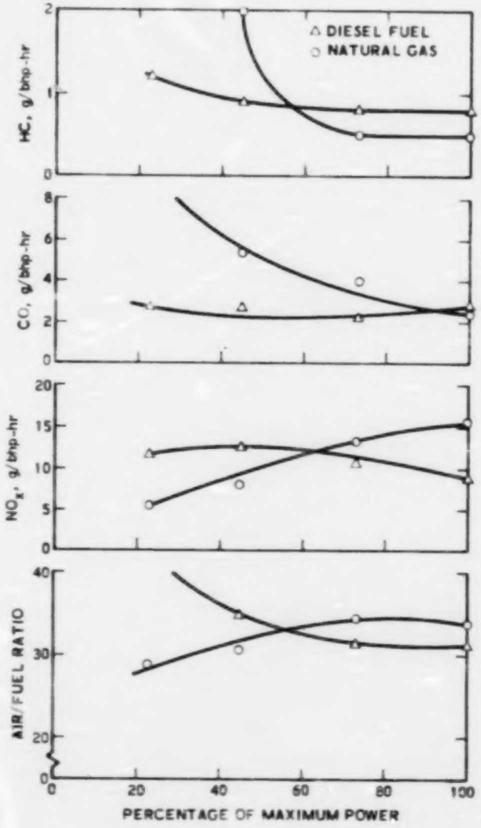


Figure 3-5. Specific mass emissions - four-stroke turbocharged, open-chamber large diesel (Engine No. 18)

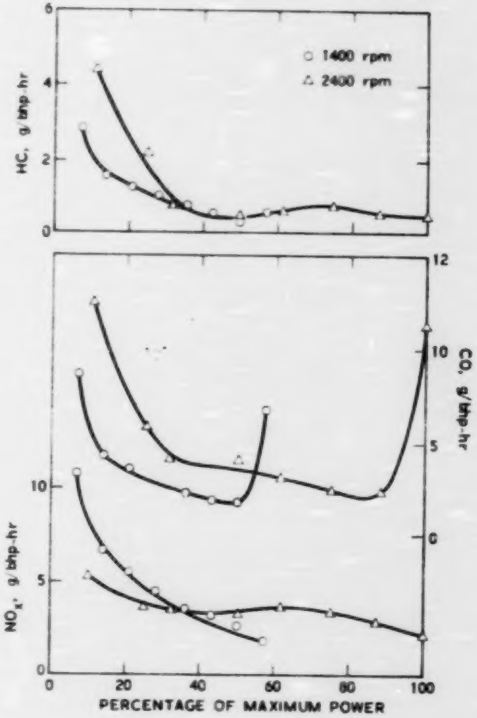


Figure 3-6. Specific mass emissions - four-stroke, naturally aspirated divided-chamber diesel (Engine No. 20)

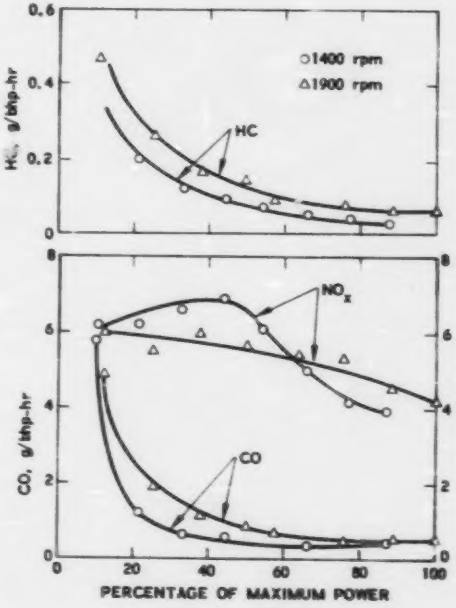


Figure 3-7. Specific mass emissions - four-stroke turbocharged, divided chamber diesel (Engine No. 23)

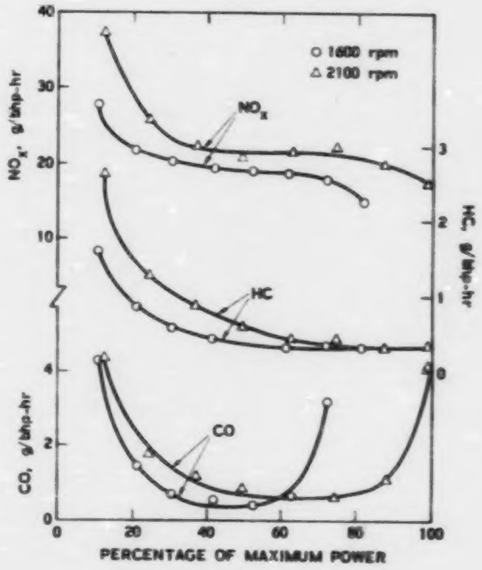


Figure 3-8. Specific mass emissions - two-stroke diesel (Engine No. 26)

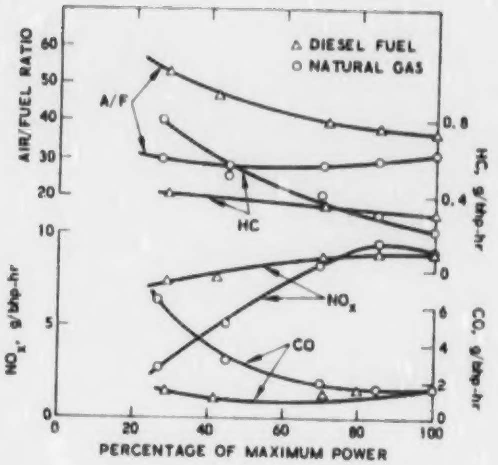


Figure 3-9. Specific mass emissions - large two-stroke, turbocharged diesel (Engine No. 30)

horsepower output obtained with decreasing load. The other naturally aspirated, open-chamber diesels evaluated exhibit similar trends, except for Engines 10 and 11 (Table 3-2) which show a small reduction in HC down to about 60 percent of full load, followed by a rather sharp rise. The CO specific mass emissions decrease with decreasing load, reaching a minimum at about 60 to 70 percent of full load and then increase again as load is further reduced. The NO$_x$ emissions of Engine No. 3 increase steadily with decreasing load, in spite of the previously noted reduction in NO$_x$ concentration. Seven of the eleven naturally aspirated, open-chamber diesel engines considered in this study show similar NO$_x$ trends. Conversely, in the remaining engines NO$_x$ decreases somewhat with decreasing load. The difference in the NO$_x$ versus load relationship might be due to differences in the fuel injection timing schedules utilized on the various engines. In some engines, the timing is retarded as load is reduced whereas others maintain a constant start of injection which in effect results in a timing advance because the injection period becomes shorter with decreasing fuel flow rate (load). As further discussed in Section 4.1.1.3, injection timing retard reduces NO_x whereas advanced timing increases NO_x.

Test data from two four-stroke turbocharged, open-chamber diesel engines are shown in Figures 3-4 and 3-5. Similar to naturally aspirated diesels, the HC and CO specific mass emissions follow the expected trends, although the rates of change appear to be lower with turbocharging. At rated speed, the NO_x specific mass emissions of Engine No. 13 remain essentially constant down to about 40 percent load but increase rapidly at lower loads. Conversely, at the intermediate speed, NO_x increases steadily with decreasing load. Engine No. 18 (Table 3-3), when operated on No. 2 diesel fuel, shows rising NO_x as the load is reduced, whereas a reduction in NO_x is realized with natural gas. The difference in the NO_x trends is attributed to differences in the engine air-fuel ratio versus load and timing versus load schedules utilized for the two fuels.

Figure 3-6 presents the emission map of a small, naturally aspirated, divided-chamber diesel engine operated at two speeds. The HC emissions remain essentially constant between full load and 40-percent load but increase sharply at load levels below 40 percent. Again, CO follows the trends of the other diesel engine categories while NO_x tends to increase with decreasing load. Engine No. 21 shows similar trends.

The specific mass emissions of a typical four-stroke, turbocharged, divided-chamber diesel engine are presented in Figure 3-7, showing steadily increasing HC as engine load is reduced. As in open-chamber, turbocharged diesels, CO varies very little in the 60- to 100-percent load regime but increases rapidly at lower loads. NO_x increases steadily with decreasing load.

Emission maps of two typical two-stroke diesel engines are presented in Figures 3-8 and 3-9. One of these is a medium-size, blower-scavenged engine operated on No. 2 diesel fuel while the other is a large, turbocharged unit which was run on diesel fuel and on natural gas. The emission characteristics of these engines are comparable to many four-stroke engines.

In general, the HC and CO emissions of the two-stroke engines are quite low, reflecting the high air-fuel ratio at rated conditions. Again, the different mass emission trends obtained with the two types of fuel are attributed to differences in the air-fuel ratio schedules used for the two fuels.

3.1.4.3.3 Selected Diesel Emission Values

Based on the analysis of the available diesel engine data discussed Sections 3.1.4.3.1 and 3.1.4.3.2 average HC, CO, and NO_x emission concentrations and specific mass emissions at rated and part-load operating conditions have been determined for the following uncontrolled engine categories.

1. Four-stroke, naturally aspirated, open-chamber
2. Four-stroke, turbocharged, open-chamber
3. Four-stroke, naturally aspirated, divided-chamber
4. Four-stroke, turbocharged, divided-chamber
5. Two-stroke, blower-scavenged
6. Two-stroke, turbocharged

The average pollutant specie concentrations and specific mass emissions at rated engine conditions are presented in Table 3-8. Where appropriate, the emissions are listed separately for the medium-size engines, most of which were originally designed for mobile installations, and the large stationary engines. Also listed in this table are the observed emission variations. As indicated in the table, the emissions from these uncontrolled engines vary over wide ranges. For example, in naturally aspirated, open-chamber engines, the HC emissions vary between 0.1 g/bhp-hr and 2.2 g/bhp-hr. In view of the magnitude of these variations it is very difficult to establish typical emissions for each engine category.

TABLE 3-8. AVERAGE DIESEL ENGINE EMISSIONS AT RATED CONDITIONS (UNCONTROLLED ENGINES)

Engine Category	Size	Average Concentrations, ppm			Average Specific Mass Emission, g/bhp-hr			Concentration Range, ppm			Specific Mass Emission Range, g/bhp-hr		
		HC	CO	NO_x	HC	CO	NO_x	HC	CO	NO_x	HC	CO	NO_x
Four-Stroke, Naturally Aspirated, Open Chamber	Medium large	485 -	1630 -	1370 -	1.0 0.5	6.5 4.2	8.9 15.2	70 - 1000 -	540 - 3300 -	740 - 2550 -	0.1 - 2.2	2.2 - 14.6 one engine only	5.3 - 16.6
Four-Stroke, Turbo-Charged, Open Chamber	Medium large[a]	575 -	680 -	1440 -	1.5 0.6	3.4 2.6	12.0 11.5	130 - 1140 -	530 - 1000 -	1150 - 1600 -	0.3 - 2.9 0.1 - 0.8	2.6 - 5.2 1.1 - 4.5	9.3 - 15.6 9.0 - 15.6
Four-Stroke, Naturally Aspirated, Divided Chamber	Small, light duty	90	1430	440	0.3	7.3	3.1	55 - 120	830 - 2025	280 - 600	0.1 - 0.3	3.3 - 9.1	2.1 - 4.0
Four-Stroke, Turbo-Charged, Divided Chamber	Medium	46	118	636	0.2	0.6	4.7	23 - 80	52 - 200	545 - 700	0.1 - 0.2	0.3 - 0.9	4.3 - 5.0
Two-Stroke, Blower Scavenged	Medium	167	906	1472	0.5	5.4	14.7	130 - 200	585 - 1135	1350 - 1675	0.4 - 0.6	3.6 - 7.1	12.5 - 17.1
Two-Stroke, Turbo-Charged[a]	Large	-	-	-	0.4	2.1	8.6	-	-	-	0.3 - 0.6	1.6 - 3.0	8.0 - 9.0

[a] With intercooling

According to Table 3-8, the divided chamber diesels show superior emission performance particularly with respect to HC and NO_x. In general, the specific mass emissions of the large, low-speed stationary diesels tend to be lower than for the medium-size units.

Part-load emission correlations for the six diesel engine categories identified above are presented in Figures 3-10 through 3-12, showing HC, CO, NO_x emission ratios at rated speed as a function of the ratio of engine power to rated power. The emission ratios are defined as the ratio of the specific mass emissions at a given power output point to the specific mass emissions at full load. The curves represent arithmetic averages of all the available data in each engine category. It should be noted that only a very limited number of data points are available for several of the engine categories and therefore the trends shown in the figures may not be representative of all engines.

As illustrated in Figure 3-10, the average HC specific mass emission ratio of all diesel engine categories increases steadily with decreasing load. The naturally-aspirated diesels show a larger rate of increase than the turbocharged and blower scavenged designs.

The CO specific mass emission ratios shown in Figure 3-11 indicate significant variations. In most cases, CO decreases rapidly as the load is reduced from the rated value, but increases again

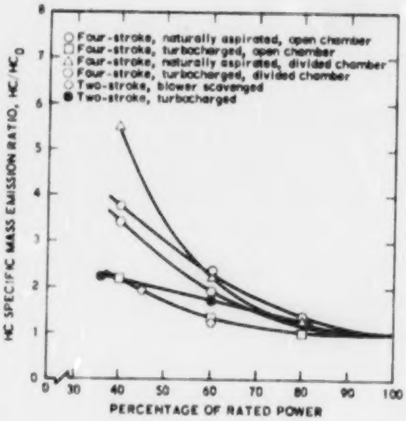


Figure 3-10. Diesel engine part-load hydrocarbon emissions - rated speed

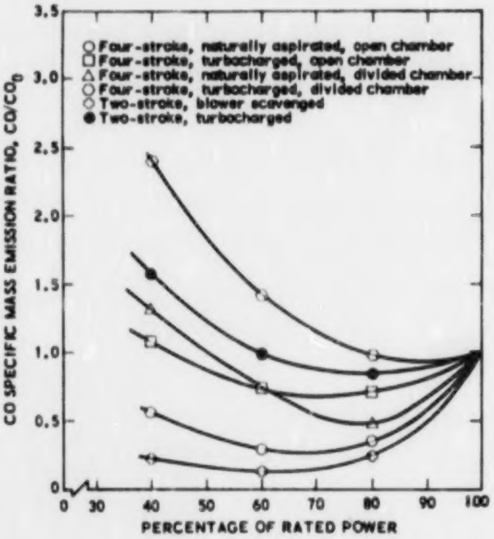


Figure 3-11. Diesel engine part-load carbon monoxide emissions - rated speed

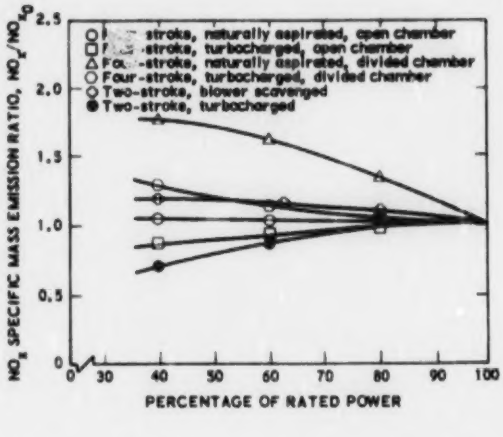


Figure 3-12. Diesel engine part-load oxides of nitrogen emissions - rated speed

at very low load levels. This initial reduction of CO is related to the associated increase in air-fuel ratio, whereas the increase at low loads is attributed to quenching effects.

The NO_x specific mass emission ratios at rated speed are presented in Figure 3-12. Except for the four-stroke, naturally aspirated divided-chamber diesel engines, the variations in NO_x in the 50 to 100 percent load regime remain within ±20 percent. However, it should be emphasized again that individual engines in some of the engine categories considered here show considerably larger variations. For example, at 40 percent load, Engine 30 when operated on natural gas shows a NO_x emission ratio of 0.52 instead of the average value of 0.7 plotted in the figure.

Somewhat different part-load emission characteristics are obtained when engine speed and load are reduced simultaneously. In this case, the HC and CO specific mass emission factors at part-load would be as much as 50 percent lower. However, for some engines this reduction would be accompanied by an increase in NO_x while for others there would be no change in NO_x or even some reduction. Although there is insufficient data available to permit the formulation of a generally applicable speed versus load correlation, it is conceivable that some reduction in NO_x could be achieved by properly derating the engine. This is further discussed in Section 4.1.1.1.

The part-load emission factors presented in Figures 3-10 through 3-12 combined with the average specific mass emissions listed in Table 3-8 for the rated engine conditions provide all the information necessary for the computation of predicted specific mass emissions at part-load.

3.1.4.4 Smoke and Particulate Emissions

3.1.4.4.1 Smoke

The smoke emitted from diesel engines can be grouped into three categories: white, blue, and black. White smoke, or cold smoke, which usually appears in the exhaust during an engine cold start, consists primarily of raw fuel combined with other compounds such as aldehydes. It is formed in the wall region of the cylinders where the temperatures are not high enough to ignite the fuel. Blue smoke, emitted by some diesels, is the result of the combustion of excessive amounts of lubricating oil. Black smoke, or hot smoke, consists of agglomerated soot particles containing very small (200-300 Angstrom) carbon particles and some hydrogen.

Black smoke is the most important specie and is formed by vapor phase pyrolysis of fuel molecules which occurs when insufficient oxygen is available in the high temperature zones of the combustion chamber (Ref. 3-24). The soot formation process depends on (1) the amount of incompletely mixed fuel and the fuel-air ratio of the mixture at the time of combustion, and (2) the associated temperatures. An increase in temperature generally increases soot release. Once formed, the soot particles must find oxygen to burn and if this is not accomplished within the confines of the combustion chamber, visible smoke appears in the exhaust (Ref. 3-25). In general, inadequate fuel atomization and mixing with the air, overfueling, and maladjustment in the fuel injector system tend to increase soot formation (Ref. 3-2). Black smoke is more likely to occur at high loads where less excess air is available for combustion. Normally, turbocharged diesels operate at higher air-fuel ratios than naturally aspirated engines, and, as a result they tend to emit less smoke. Also, high-pressure fuel injection systems are believed to contribute to lower smoke levels (Ref. 3-26).

Smoke data from four naturally aspirated, four-stroke diesel engines are depicted in Figure 3-13 (Ref. 3-12). The sharp increase in the smoke level of Engines 1 and 9 (designated Engines B and A in Ref. 3-12) at high loads may be caused by poor injector

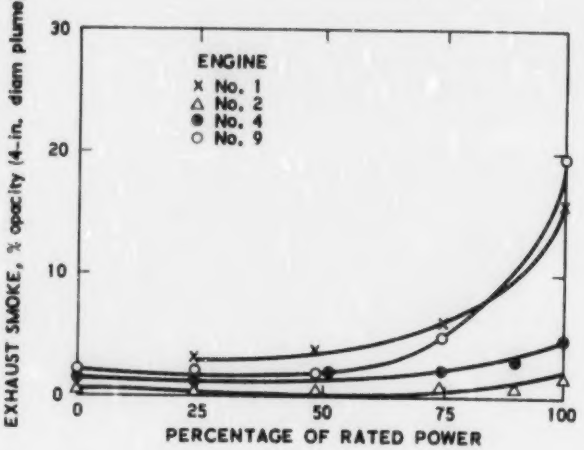


Figure 3-13. Exhaust smoke vs output power for four open-chamber, naturally aspirated diesel engines (Ref. 3-12)

performance resulting in inadequate fuel atomization and fuel impingement on the cylinder walls. As shown in Tables 3-2 and 3-3, the average smoke opacity of naturally aspirated diesel engines at rated load is about 13 percent, while the turbocharged engines show only about three-percent opacity. There is insufficient data to characterize the smoke emissions of the other diesel engine categories. However, it is believed that little smoke should be emitted by the two-stroke engines because of the relatively high air-fuel ratio utilized in these engines under all operating conditions.

Average steady-state smoke intensity data reported by Southwest Research Institute for a number of naturally-aspirated and turbocharged, open chamber and prechamber diesel engines are listed in Table 3-9 (Ref. 3-16). The data show considerable engine-to-engine variability. In all cases, the highest smoke levels were obtained at full load and intermediate speed. Under these conditions the air-fuel ratio is generally at its minimum value. The high, full-load smoke levels of the two turbocharged open-chamber diesels at the intermediate speed condition are attributed to operation near the stoichiometric air-fuel ratio.

TABLE 3-9. AVERAGE STEADY-STATE SMOKE EMISSION FROM DIESEL ENGINES (Ref. 3-16)

Engine	Chamber Type	Aspiration	Smoke Intensity in % Opacity at Condition						
			Low Idle	Load at Intermediate Speed			Load at Rate Speed		
				0	Half	Full	0	Half	Full
International D407	Open	Natural	1.2	1.0	4.5	20.0	1.0	4.0	9.2
Perkins 4.236	Open	Natural	1.0	1.0	1.0	10.3	1.7	1.4	7.7
Allis Chalmers 3500	Open	Turbo	0.5	1.0	2.8	31.5	1.2	5.0	7.3
John Deere 6404	Open	Turbo	2.0	1.9	7.3	25.5	2.2	5.5	6.0
Mercedes OM636	Divided	Natural	1.0	1.5	1.5	8.5	2.0	1.5	8.0
Onan DJBA	Divided	Natural	0.5	0.8	2.0	2.5	1.0	1.0	3.0
Caterpillar D6C	Divided	Turbo	2.3	2.0	3.0	4.5	3.2	2.3	2.7
Detroit Diesel 6V-71	Open	Blower	0.5	0.8	1.0	1.0	1.0	1.0	1.5

TABLE 3-10. AVERAGE PARTICULATE EMISSIONS FROM DIESEL ENGINES (Ref. 3-16)

Engine	Chamber Type	Aspiration	Particulate Concentration, mg/sfc[a]						
			Low Idle	Load at Intermediate Speed			Load at Rate Speed		
				0	Half	Full	0	Half	Full
International D407	Open	Natural	4.5	4.3	8.1	20.0	6.1	7.1	12.0
Perkins 4.236	Open	Natural	1.1	1.3	1.2	11.6	9.8	9.3	10.0
Allis Chalmers 3500	Open	Turbo	5.2	3.6	4.0	17.4	2.6	3.4	3.6
John Deere 6404	Open	Turbo	4.5	7.2	7.2	22.4	3.7	3.8	6.2
Mercedes OM636	Divided	Natural	3.1	3.4	5.6	9.4	8.9	7.9	8.7
Onan DJBA	Divided	Natural	6.4	4.5	3.6	5.2	5.5	8.6	8.1
Caterpillar D6C	Divided	Turbo	2.4	0.9	1.2	1.6	0.8	2.0	2.4
Detroit Diesel 6V-71	Open	Turbo	0.9	0.3	0.6	0.6	0.2	0.6	0.7

[a] Standard cubic foot

Smoke (and particulate) emissions from diesel engines can be minimized, but not eliminated, by several means including engine derating, proper fuel injection system design and maintenance, and use of smoke suppressant fuel additives. The latter two approaches are further discussed in Sections 4.1.1.3 and 4.1.1.4. Some reduction in smoke has apparently been achieved by means of intake air heating (Ref. 3-27).

3.1.4.4.2 Particulates

Particulate emission data from eight industrial diesel engines tested by Southwest Research Institute are presented in Table 3-10. Comparison with the smoke data of Table 3-9 indicates that the particulate levels of these engines correlate to some degree with visible smoke, particularly at high smoke levels. However, in some cases a considerable amount of particulate matter was measured under conditions where smoke was barely readable. High smoke levels were always associated with high particulate emissions (Ref. 3-16). Data from another source indicate that a correlation might exist between the opacity of black smoke and the mass density of soot in the exhaust gas (Ref. 3-28). Tests conducted on a Mercedes Benz light-duty 220D diesel engine indicate that the particulate emissions of this engine are of the order of 2 to 3 times higher than those obtained from equivalent gasoline engines using leaded fuel and about 10 to 20 times higher than for non-leaded fuel. The material collected in these tests was pitch black and very fine (Ref. 3-29). Similar results were reported by Daimler Benz (Ref. 3-30).

3.1.4.5 Odor

Diesel exhaust odor has long been recognized to be a very undesirable exhaust emission product. Although a considerable amount of research has been under way for some time, progress toward determination of the cause of the odor has been very slow because of the complexity of the heterogeneous combustion occurring in diesels and the lack of sufficiently accurate instrumentation

(Ref. 3-10). The diesel odor is believed to be related to the mixing and combustion process, the chamber shape and, to a lesser degree, fuel type and fuel composition. Most likely, the odorants are formed in the wall quench layer of the combustion chamber and in the heterogeneous pockets of unflammable fuel-air mixture. (Ref. 3-31.)

In general, diesel engine exhaust includes small amounts of unburned fuel products of partial oxidation and pyrolysis products (Ref. 3-31). Some of these compounds have strong odors, even in very low concentrations, and mixtures of these materials are believed to be responsible for the formation of the typical diesel odor (Ref. 3-2). Two major types of odor have been identified, one being an "oily-kerosene" type apparently caused by aromatic hydrocarbons and the other being of the "smoky-burnt" type, and probably caused by partially oxidized compounds. Barnes (Ref. 3-32) has concluded that diesel odor is formed by partial oxidation reactions in ultra-lean regions of the engine which are inevitably present in heterogeneous combustion as characteristic of diesel engines.

In the absence of reliable instrumentation, odor has been evaluated in the past by specially selected human panels trained to recognize both odor quality and intensity (Ref. 3-33 through 3-35). The panel classifies the odor by comparing it to twenty-eight different odor qualities and intensities supplied from the bottles of the quality/intensity (Q/I) evaluation kit. The overall composite rating "D" ranges from D-1 to D-12, with D-12 representing the strongest odor. Burnt-smoky quality "B", aromatic quality "A", oily quality "O", and pungent quality "P", have a range between 1 and 4.

More recently, a diesel odor analysis instrument has been developed by A. D. Little, Inc. on a Coordinating Research Council (CRC) contract which is designed to permit direct measurement of diesel odor levels (Ref. 3-36). Evaluation of the instrument is in progress.

Formulation of empirical correlations between odor intensity and the concentrations of other pollutants has been attempted by many investigators. Although a fairly good correlation between odor and total HC emissions has been found in some engines, correlations of this type have failed to hold for others (Ref. 3-12 and 3-37). Attempts to correlate odor with CO and NO_x emissions have failed entirely (Ref. 3-37). Tests conducted by Vogh (Ref. 3-38) indicate that light molecular weight aldehydes have very little effect on odor formation in diesel engines, and of the oxides of sulfur and nitrogen, only nitrogen dioxide represents a potential contributor to odor.

The effect of engine operating condition on odor intensity can vary markedly for different engines. In naturally aspirated engines operated at rated speed, the minimum odor intensity seems to occur in the mid-power range of the engine; whereas in turbocharged engines the odor intensity remains essentially constant over the power range (Ref. 3-12). This trend is believed to be the result of a reduction in the ignition delay time occurring at higher power levels due to an increase in the intake manifold and compression temperatures. Since odorants are thought to be formed by partial oxidation of the fuel prior to ignition, the constant odor intensity pattern would follow from this effect (Ref. 3-12). Somewhat higher odor levels were reported by Springer and Hare (Ref. 3-33) for a two-stroke engine-powered diesel bus which was tested over a range of operating conditions. The odor was most severe during acceleration and idle. This vehicle was also tested with a catalytic converter installed in the exhaust which resulted in some reduction in the odor intensity. This is further discussed in Section 4.1.3.3.

Tests conducted by Caterpillar indicate that the odor level from prechamber diesels can be 60 to 80 percent lower than for open chamber engines (Ref. 3-25). Apparently, the odor notes from both engine types are similar when the same fuel is used.

Several approaches aimed at reducing diesel odor have been considered by industry including injector modifications, catalytic mufflers and fuel additives. Utilization of a needle valve injector with a very small dribble volume in place of a check valve type injector has resulted in a substantial reduction of the odor intensity and aldehyde emissions from General Motors two-cycle GV-71 diesels (Ref. 3-35 and 3-39). With the new injectors, the aldehyde emissions were reduced by 50 to 85 percent over the operating range of the engine. Since the odor intensity varied only slightly at these different operating points, it has been concluded by General Motors that aldehydes might not be the main contributors to diesel odor.

3.1.4.6 Noise

Diesel engines are inherently noisier than gasoline engines, particularly open chamber designs most frequently used in heavy duty automotive and stationary applications. The noise is especially pronounced at idle and during engine cold start.

The principal forces contributing to diesel noise are created by the combustion pressure transients, piston slap, and timing gear impacts, with valve gear and fuel injection system impacts being lesser sources (Ref. 3-40 and 3-41). The major noise emitting engine surfaces include the valve covers, intake manifold, gear cover, and oil pan (Ref. 3-26).

Engine noise is a systems problem and reduction can be attempted either at the source or by reducing the vibration levels of the external surfaces of the engine (Ref. 3-40). Minimization of the combustion pressure rise rate results in substantial noise abatement, as evidenced by the favorable noise characteristics of prechamber and swirl-chamber diesels. The noise contributed by the piston and timing gear slap might be diminished to some degree by reducing certain component clearances within the constraints dictated by the design and manufacturing techniques (Ref. 3-41).

Further reduction in engine noise might be achieved by means of structural modifications on the engine. These include: (1) higher stiffness of the basic engine structure to reduce vibration,

(2) isolation of non-structural members, such as covers and accessories, (3) damping of certain engine components to modify their vibration characteristics and (4) application of acoustic enclosures around the engine (Ref. 3-42). Vibration isolation and stiffening have resulted in a reduction of the diesel noise by about 5 dB(A) while total enclosure of the engine has reduced the noise level by as much as 20 dB(A) (Ref. 3-43). Also, sheetmetal shields fitted with fiber glass liners have been used to cut diesel noise by as much as 4 dB(A) (Ref. 3-26).

3.1.5 Fuel Consumption

Typical performance curves for naturally-aspirated and turbocharged, open-chamber diesel engines are presented in Figure 3-14, showing engine torque, brake horsepower and specific fuel consumption as a function of operating speed (Ref. 3-44). The data are from a large stationary engine but the indicated performance trends apply to smaller diesels and divided chamber configurations as well. As indicated in the figure, the specific fuel consumption of naturally-aspirated diesels has a minimum in the mid-power/mid-speed regime. Conversely, turbocharged engines operate most economically at or near full load, and fuel economy generally worsens with decreasing speed. Comparison of the data indicates that the specific fuel consumption of turbocharged diesels is slightly lower (up to 5 percent) than for naturally aspirated designs, particularly at rated conditions.

In general, the specific fuel consumption of divided-chamber diesel engines is several percent higher than that of equivalent open chamber designs. This difference is attributed to the higher surface to volume ratio of the divided chamber concept and the higher heat losses through the chamber wall. One manufacturer has stated that in new engines there is a 5 to 10 percent difference in specific fuel consumption between the two engine types. However, in the field the difference is generally less because injection system fouling tends to occur less frequently in divided-chamber engines compared to some

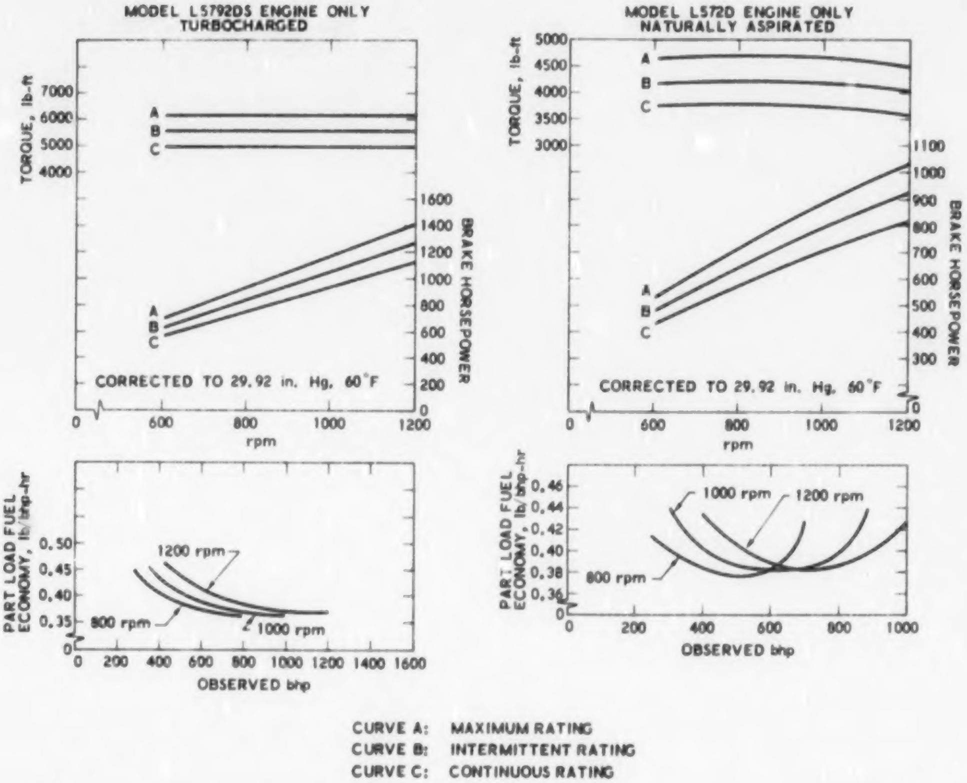


Figure 3-14. Performance of large naturally aspirated and turbocharged diesel engines (Ref. 3-44).

open chamber engines. This point is further illustrated by comparing the data listed in Tables 3-3 and 3-5 for turbocharged, open-chamber, and divided-chamber diesels, respectively. The average specific fuel consumption of the medium-size, open-chamber diesels at rated conditions is about 0.403 lb/bhp-hr compared to 0.408 lb/bhp-hr for divided chamber engines.

The naturally aspirated, divided-chamber diesel engines listed in Table 3-4 show rather high specific fuel consumption values at rated conditions which is attributed to the small size of these particular engines.

Comparison of the medium-size, two-stroke blower-scavenged diesels, (Table 3-6) and the four-stroke, naturally aspirated engines of Table 3-2 indicates slightly higher fuel consumption for the two-stroke designs. Conversely, the large two-stroke turbocharged diesels listed in Table 3-7 have a slightly lower specific fuel consumption than shown in Table 3-3 for one large four-stroke, turbocharged diesel engine.

3.2 SPARK-IGNITION ENGINE CHARACTERISTICS

3.2.1 Engine Description

The spark-ignition internal combustion engine is the most-used powerplant in the world today, and many different designs have been developed since this engine type was invented almost a century ago. In size, these engines range from small single-cylinder units producing less than one horsepower to large multicylinder units with power output ratings of several thousand horsepower. (Units above several hundred horsepower are predominantly used in stationary power applications.)

One of the basic advantages of reciprocating internal-combustion engines is the high maximum thermodynamic cycle temperature resulting in high thermal efficiency. Relative to current diesel engines, spark-ignition engines operate at lower peak pressures. Thus, the structural stresses are lower, permitting higher power-to-weight and power-to-volume ratios for spark-ignition engines.

Conventional automotive spark-ignition engines generally operate on gasoline and in certain applications on gaseous fuels such as liquified petroleum gas (LPG). Conversely, many of the stationary engines are gas fueled using natural gas, waste gas, water gas, and methane. Based on the type of work cycle employed, these engines can be divided into two- and four-stroke designs. These engine classes can be subdivided into different categories depending upon the type of fuel supply system (carburetion, fuel manifold injection, direct cylinder injection) and air supply system, (supercharged, turbocharged, naturally aspirated, and blower-scavenged) utilized.

More recently, the introduction of rotary engines (Wankel engine) has added another type of spark-ignition engine presently available for use in automotive applications. To date, this engine type has not been introduced into the stationary engine market.

Since the early 1960s, several organizations have been involved in the development of stratified charge spark-ignition engines, primarily because of their low exhaust emission, specific fuel consumption, and multifuel capabilities. The stratified charge engines can be divided into two categories: open-chamber (or direct-injection engines) and divided-chamber (or prechamber engines). These engine configurations are discussed in Section 4.2.2.4.

In the selection of a stationary spark-ignition engine, the emphasis is on investment cost, economy of operation, reliability, and durability. Most of the stationary engines operate at semi-constant speed and load conditions. This permits incorporation of many design simplifications (simple carburetion, fuel injection, spark timing, and related control systems), as well as optimization of important engine operating and design parameters (valve timing and porting, manifolding, etc.). Generally, little or no weight or space limitations are imposed on stationary engines, resulting in greater freedom in the selection of materials and in the design of certain engine accessories (e.g., cooling system).

In regard to exhaust emission control, stationary engines have distinct advantages over mobile engines. For instance, because of the semisteady state operation of these engines, many design and operating parameters could be optimized for exhaust emissions. Furthermore, because of the freedom from space and weight limitations, various exhaust gas purification systems could be applied, which would be impractical for use in mobile installations.

3.2.2 Applications

Stationary spark-ignition, internal-combustion engines are used in a great variety of industrial, municipal, and urban applications. Small gasoline engines (1 to 10 HP) are utilized to drive domestic, agricultural, and commercial power tools and equipment — such as power saws, lawn mowers, and portable compressor, pump, and electric generator units. Medium-size gasoline engines (50 to 200 HP) are mostly used for commercial and construction site compressors, pumps, blowers, and electric power generator units. Medium-large spark-ignition engines (200 to 1000 HP) are generally operated on gaseous fuels and are used for heavy-duty, medium-speed applications. Most of them are of the four-stroke, naturally aspirated type and are used to power gas compressors or stand-by power generators. Large spark-ignition engines (1000 HP and up) are always operated on gaseous fuels and are both four- and two-stroke, low-speed (300 to 400 rpm) engines. The application of these engines includes compressor drives, gas-well recompression (in transmission lines), gas plant compressors, refinery process compressors, water pumping, sewage pumping, and electric power generator drives for continuous operation. Table 3-11 presents a breakdown of the stationary spark-ignition engines installed during the 1963-1970 time period (Ref. 3-9). The total number of gasoline and gas-operated spark-ignition engines in use at the end of 1970 outnumbered the diesel and gas-turbine engines by almost two orders of magnitude.

3.2.2.1 Installed Power

Table 3-12 presents the breakdown of the estimated installed horsepower of spark-ignition gas engines at the end of 1971 (Ref. 3-9). Because of the great diversity of gasoline engine applications and sales to distributors rather than directly to the customers, there is insufficient information available at this time to provide a

TABLE 3-11. INTERNAL COMBUSTION ENGINES – NUMBER VS END USE (Ref. 3-9)

Year	1970	1969	1968	1967	1966	1965	1964	1963	1963-70 Cumulative Total
Gasoline									
Marine	61,663	106,693	84,624	103,478	103,899	39,937	29,463	28,005	557,762
Lawn & Garden	8,013,961	8,717,864	8,236,693	7,555,701	6,422,221	5,766,819	4,760,683	5,084,262	56,291,668
Chain Saws					482,194	442,855	435,152	373,263	
Agriculture	169,977	187,437	193,380	202,167	509,543	417,507	434,175	430,362	2,544,548
Subtotal	8,245,601	9,011,994	8,514,697	7,861,346	7,517,857	6,667,118	5,659,473	5,915,892	59,393,978
Construction	152,178	175,605	104,638	121,225	132,214	85,076	66,052	51,136	888,124
Generator Sets	86,264	90,760	67,798	67,930	76,678	67,769	59,190	43,542	559,931
General Industrial	1,073,564	1,249,185	1,134,638	1,070,887	1,173,939	1,087,760	949,007	851,068	8,590,048
Subtotal	1,312,006	1,515,550	1,307,074	1,260,042	1,382,831	1,240,605	1,074,249	945,746	10,038,103
Total	9,557,607	10,527,544	9,821,771	9,121,388	8,900,488	7,907,723	6,733,722	6,825,638	69,432,081
Gaseous Fuels									
Agriculture	2,987[a]	3,257	3,947	6,873	11,460	5,654	5,780	8,788	46,746
Construction Generator Sets	2,342[a]	2,694	3,547	4,799	5,539	8,266	7,700	5,142	40,029
General Industrial	1,821	1,002	1,028	1,260	1,135	839	911	562	8,558
Total	7,150	6,953	8,522	12,932	18,134	14,759	14,391	14,492	97,333

[a] Estimates based on distribution of gas engines in Agriculture and Construction/Generator Sets for previous three years

TABLE 3-12. ESTIMATED INSTALLED HORSEPOWER OF SPARK-IGNITION GAS ENGINES FOR 1971[a]

Application	Installed Horsepower
Electric Power Generation	90,000[b]
Oil and Gas Pipelines	10,990,000
Natural Gas Processing Plants	2,410,000
Oil and Gas Exploration	500,000
Crude Oil Production	852,000
Natural Gas Production	3,237,000
Industrial Process	230,000
Municipal Water and Sewage	465,000
Total	18,774,000

[a] A summary table listing the installed horsepower of all stationary engines is presented in Appendix A

[b] Estimated 1970 data

breakdown of installed gasoline engine horsepower. The total installed horsepower of small gasoline engines (1 to 10 bhp) is estimated to be about 180 x 10^6 hp and of the medium size gasoline engines (25 to 250 bhp) to about 600 x 10^6 hp. This brings the grand total of all stationary spark-ignition engine (including gas engines) installed horsepower to about 800 million.

3.2.2.2 Operating Modes

The duty cycles of large, low-speed spark-ignition engines, driving electric power generators, industrial, agricultural and municipal water and sewage pumps, compressors, etc., are similar to the operating modes of diesels which are discussed in Section 3.1.3.2. The duty cycle and the application of small- and medium-size gasoline spark-ignition engines varies widely. Unlike light-duty automotive gasoline engines, the majority of the small- and medium-size stationary gasoline engines operate at constant or nearly constant speed, with power output controlled by a governor.

3.2.2.3 Fuel Requirements

Stationary spark ignition engines can be operated on various fuels including leaded, low-lead, and nonleaded gasoline; white gas; gaseous fuels and various blends of other hydrocarbone fuels. In most stationary engines the fuel octane requirement is quite low.

Commercially available gasoline blends have a low sulfur content (0.025 to 0.048 percent by weight) and contain insignificant amounts of other contaminants. However, gum and carbon deposits in the induction system, combustion chamber, and crankcase have a tendency to accumulate, particularly for those duty cycles which involve frequent engine starts and extended idle periods.

In general, gaseous fuels cause less deposit build-up in the combustion chamber — which is attributed to lean air-fuel ratio operation. This inherent advantage is reflected in better brake specific fuel consumption and lower hydrocarbon and carbon monoxide emissions.

3.2.3 Emissions

Over the past two decades, extensive research has been conducted in the field of spark-ignition engine emissions. In general, the research activities have been concerned with automotive engines; however, many of the findings and conclusions are applicable to stationary spark-ignition engines as well.

3.2.3.1 General

Uncontrolled spark ignition engines emit air pollutants from four sources: engine exhaust, crankcase blowby, carburetor, and fuel tank. According to a survey conducted on a large number of uncontrolled automotive spark-ignition gasoline engines, (Ref. 3-45), the engine exhaust contributes 100 percent of the carbon monoxide (CO) and oxides of nitrogen (NO) emissions and about 65 percent of the total unburned hydrocarbons (HC). Another 25 percent of the HC is attributed to crankcase blowby, with an additional five percent each resulting from evaporation of gasoline from the carburetor and from the fuel tank. Furthermore, particulates constituting approximately 5 percent by weight of the unburned hydrocarbons are emitted from the engine exhaust (Ref. 3-46). These consist of lead compounds, carbon particles, motor oil, and nonvolatile reaction products formed during the combustion process in the engine cylinder. There are literally hundreds of reaction products emitted — most of them in trace quantities — including organic acids, high molecular weight olefins, carbonyl compounds, and sulfur compounds whose concentration depends on the sulfur content of the gasoline.

Statistical data on the emissions from spark-ignition engines operated on gaseous fuels (e.g., natural gas) are currently not available. However, in this case the evaporative emissions are essentially zero since the fuel system is not vented.

3.2.3.2 Principles of Engine Combustion-Generated Emissions

The combustion process in spark-ignition engines involves complex chemical reactions, whose "kinetics" depend on the chemical composition and the thermodynamic state of the reactants (fuel and air) and on the parameters of the reaction vessel (e.g., the engine combustion chamber) which affect the combustion process. Since the nitrogen contained in the combustion air is heated to high temperatures during the combustion process, some of it reacts with oxygen to produce nitric oxide (Ref. 3-47). As the combustion gases are rapidly cooled during their expansion and exhaust, the nitric oxide remains in a state of "frozen equilibrium", i.e., it does not dissociate to nitrogen and oxygen as predicted from chemical equilibrium considerations (Ref. 3-48). In gasoline engines combustion occurs frequently in rich fuel-air mixture zones resulting in incomplete oxidation of the fuel and production of significant quantities of CO (Ref. 3-49). This is further compounded by flame-quenching in the wall region of the combustion chamber. As a result, part of the fuel in the wall boundary layer remains uncombusted and is then expelled during the exhaust stroke (Ref. 3-50).

The exhaust emissions from spark-ignition engines are primarily affected by the air-fuel ratio of the combustible mixture. At air-fuel ratios below stoichiometric, little NO is produced because of a lack of available oxygen. However, under these conditions, substantial quantities of CO are formed and a portion of the excess fuel is converted to organic compounds, collectively called unburned hydrocarbons.

The concentration of nitric oxide reaches its peak at air-fuel ratios slightly leaner than stoichiometric. Further leaning of the mixture results in decreasing concentration of NO, because of the attendant decrease in the peak combustion temperature (Ref. 3-51). CO reaches the minimum concentration at air-fuel ratios above about 15.5 and HC continues to decrease with leaning of the mixture until misfire sets in. At that point, the HC emissions increase again.

Another variable which has a significant effect on the concentration of the exhaust pollutants is the combustion temperature. The theoretical concentration of nitric oxide is an exponential function of the combustion temperature. Conversely, the CO concentration is determined primarily by the air-fuel ratio and is only slightly affected by the combustion temperature (Ref. 3-52). The HC emissions are affected by the combustion temperature only insofar as lowering the combution temperature can extend the burning time of the mixture resulting in higher exhaust temperatures and more complete oxidation of the HC species in the exhaust manifold (Ref. 3-53).

In general, all spark-ignition engine parameters affect the emissions inasmuch as they affect either the air-fuel ratio or the combustion temperature, or both (a detailed discussion of these aspects is presented in Section 4.2.1). In addition, test data indicate that combustion chamber wall effects and deposits might have some effect on the emissions of NO_x and HC (Ref. 3-54 through 3-57).

When a spark-ignition engine is operated on gasoline at substantially leaner than stoichiometric air-fuel ratio, a distinct aldehyde odor in the exhaust is sometimes noticeable. This may result from partial oxidation of fuel hydrocarbons in ultra-lean zones in the combustion chamber. As mentioned in Section 3.2.3.1, particulates, consisting of lead compounds and carbon particles, are emitted from the exhaust of spark-ignition engines operated on commercial gasolines. However, with gaseous fuels (e.g., natural gas), the particulates are virtually eliminated and the exhaust odor is substantially improved (Ref. 3-58).

3.2.3.3 Test Procedures and Instrumentation

The test procedures and instrumentation for determination of exhaust emissions from mobile and stationary internal combustion engines are described in detail in numerous publications (Ref. 3-59 through 3-68). Table 3-13 presents an overview of the principles, methods, and instruments generally used for the detection and analysis of the various exhaust pollutants of spark-ignition engines.

TABLE 3-13. EXHAUST GAS ANALYSIS METHODS AND INSTRUMENTS

Specie Detected	Method	Type of Method	Remarks	Ref.
NO, NO_2	Phenoldisulfonic (PDS)	Wet chemical	Grab samples Analysis time 24 hr	3-59
NO, NO_2	Saltzman	Wet chemical	Grab samples Analysis time 1 hr	3-60
NO	Nondispersive Infrared (NDIR)	Optical	Continuous analysis Sensitized to NO	3-68
NO	Chemiluminescence (CL)	Chemiluminescence	Continuous analysis, response 2 sec, high sensitivity	3-61
HC	Nondispersive Infrared (NDIR)	Optical	Continuous analysis Sensitized to n-hexane	3-62
HC	Flame Ionization Detector (FID)	Flame Ionization	Continuous analysis Total HC detection	3-63
HC	Gaschromatographic (GC)	Chromatographic	Grab samples Individual HC components	3-64
CO	Nondispersive Infrared (NDIR)	Optical	Continuous analysis Sensitized to CO	3-62
CO_2	Nondispersive Infrared (NDIR)	Optical	Continuous analysis Sensitized to CO_2	3-62
Aldehydes	Gaschromatographic (GC)	Chemical and chromatographic	Grab samples, quantitative and qualitative analysis	3-65
Particulates	Scanning electron microscope (SEM)	Electron microscope X-Ray spectrometer	Grab samples Resolution 200Å	3-66

As the methods and instruments are perfected and the understanding of the importance of particular pollutants in atmospheric photochemistry is improved, the test procedures are subject to progressive modifications or improvements. The officially adopted test procedures and the approved instruments for exhaust analysis are described in the Federal Register (Ref. 3-68). Moreover, the test procedures for diesel engines, presented in Section 3.1.4.2, are applicable to stationary spark-ignition engines as well.

3.2.3.4 Gaseous Emissions

The characteristics of exhaust emissions of NO_x, HC, and CO from spark-ignition engines are strongly influenced by the type of fuel utilized. The influence of liquid fuels, such as the commercial gasolines, is distinctly reflected in the composition and photochemical reactivity of the exhaust hydrocarbons (see Section 4.2.1). The gaseous fuels, on the other hand, are generally combusted with excess air, and as a result the HC emissions (and their reactivity) are quite low. The exhaust emissions are also adversely affected by fuel maldistribution in the individual engine cylinders particularly in the case of liquid fuels.

As previously noted, most (gasoline) engines are operated at rich or near-stoichiometric air-fuel ratios, resulting in high CO, HC, and NO_x emissions.

In the case of gaseous fuel (e.g., natural gas) the engine is generally operated with excess air resulting in relatively low specific mass emission of CO and HC. These inherent differences in the operating conditions of liquid- and gas-fueled spark-ignition engines are primarily responsible for the observed difference in the emissions of these two classes.

3.2.3.4.1 Emissions at Rated Conditions

The emissions from eight heavy-duty gasoline engines and from seven medium large and large gas engines are presented in Tables 3-14 and 3-15, respectively. The emissions of the gasoline engines are presented for steady-state operating conditions (estimated rated conditions for stationary applications) and in terms of composite values over the 23-mode test cycle (Ref. 3-69). The emissions from large gas engines are presented for five engines at rated operating conditions and for two engines as the composite values of the Diesel Engine Manufacturers Association (DEMA) test cycle.

From comparison of the emission data, it is apparent that the HC and CO specific mass emissions from gasoline engines are an order of magnitude higher than those of gas engines. Conversely, the NO_x emissions are consistently lower. The variation of the emissions among the gasoline engines appears to be lower than the variation among the gas engines. As indicated, the turbocharged engines show consistently higher NO_x emissions.

The emission levels of the heavy-duty gasoline engines are consistent with the emission levels of pre-controlled automotive engines. However, Tables 3-14 and 3-15 present only a small data sample and, depending on the age and mechanical condition of the engines, greater variations in the emission levels are expected to exist among the stationary engines currently in use.

3.2.3.4.2 Part Load Emissions

Figure 3-15 (from Ref. 3-70) shows the exhaust concentration of CO, HC, and NO from an automotive spark-ignition gasoline engine in the form of engine performance maps. At certain part-load conditions, minimum concentration of CO, HC, and NO results in the exhaust gas; as expected, the minimum concentration of all three pollutants occurs at distinctly different part-load operating conditions of the engine.

TABLE 3-14. EMISSIONS FROM FOUR-STROKE, NATURALLY ASPIRATED SPARK IGNITION HEAVY DUTY GASOLINE ENGINES

Engine Type and Maximum bhp/rpm	Engine Identification No.	CID	No. of Cylinders	Continuous Duty							Composite Values[b]				Reference
				Rated Power, bhp	Rated Speed, rpm	Specific Mass Emissions, g/bhp-hr			BSFC,[a] $\frac{lb}{bhp\text{-}hr}$	$\frac{Air}{Fuel}$ Ratio	Specific Mass Emissions, g/bhp-hr			BSFC,[a] lb bhp-hr	
						HC	CO	NO_x			HC	CO	NO_x		
Straight 6, 132/3500	HD 1-1	-	6	69	2300	0.37	7.6	15.8	0.531	16.0	9.20	46.2	13.9	0.677	3-69
Straight 6, 110/3800	HD 2-1	-	6	58	2300	1.74	25.1	13.1	0.548	14.7	10.05	47.8	11.9	0.750	3-69
V-8, 217/3200	HD 1-2	-	8	122	2300	1.29	10.5	13.3	0.509	15.6	12.85	46.8	12.8	0.675	3-69
V-8, 230/4000	HD 2-2	-	8	125	2300	4.02	49.5	8.7	0.501	14.7	12.61	89.8	7.9	0.756	3-69
V-8, 152/3600	HD 1-3	-	8	82	2300	0.63	7.8	13.1	0.578	15.2	13.10	35.8	11.4	0.730	3-69
V-8, 175/4000	HD 2-3	-	8	95	2300	2.76	35.5	10.6	0.503	14.4	9.06	30.4	10.7	0.631	3-69
V-8, 150/4000	22	318	8	73.8	2300	2.36	20.4	11.87	0.498	14.8	3.0	45.2	10.3	0.577	3-71
V-8, 158/4000	23	345	8	76.7	2300	1.97	28.66	11.45	0.496	14.3	2.41	31.0	9.38	0.643	3-71

[a] Brake Specific Fuel Consumption

[b] 23-mode test cycle

TABLE 3-15. EMISSIONS FROM FOUR- AND TWO-STROKE, ASPIRATED AND TURBOCHARGED SPARK-IGNITION GAS ENGINES

Engine Type	Engine Identification No.	CID	No. of Cylinders	Continuous Duty							Composite Values[b]		
				Rated Power, bhp	Rated Speed, rpm	Specific Mass Emissions, g/bhp-hr			BSFC,[a] Btu/bhp-hr	Air/Fuel Ratio	Specific Mass Emissions, g/bhp-hr		
						HC	CO	NO_x			HC	CO	NO_x
Four-Stroke, Aspirated	-	9896	12	1200	900	-	-	-	8400	-	1.32	5.84	14.2
Four-Stroke, Turbocharged	-	6597	8	1100	900	-	-	-	7600	-	7.37	0.65	19.7
Two-Stroke, Turbocharged	GMVH-8	-	8	1600	330	1.58	0.17	20.1	6632	-	-	-	-
Two-Stroke, Aspirated	GMVA-8	-	8	2000	250	1.9	0.32	15.0	7127	24.8	-	-	-
Four-Stroke, Aspirated	-	-	-	-	-	0.25	0.2	32.0	8000	21.5	-	-	-
Four-Stroke, Aspirated	-	-	-	-	-	1.25	0.6	18.0	7400	-	-	-	-
Four-Stroke, Aspirated	-	-	-	-	-	0.37	0.2	26.5	8100	21.0	-	-	-

[a] Brake Specific Fuel Consumption

[b] Composite emission values by DEMA test cycle

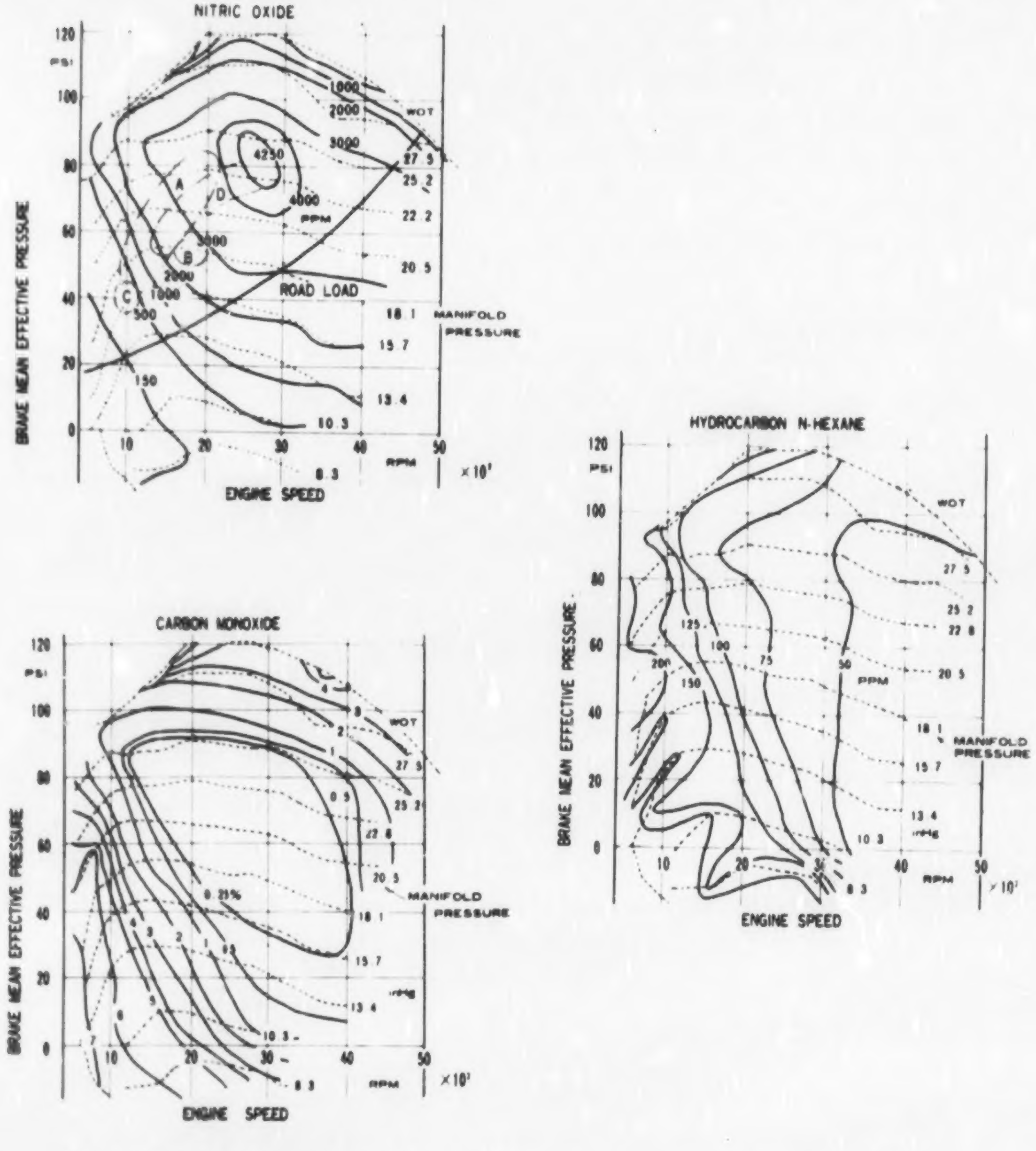


Figure 3-15. Emission map - automotive spark-ignition engine (Ref. 3-70)

Figure 3-16 presents the relative specific mass emissions of NO_x, HC, and CO, specific fuel consumption, and air-fuel ratio as a function of the percent maximum power of a heavy-duty gasoline engine (Ref. 3-71). At 2300 rpm, the specific emission of NO_x increases as the engine load is decreased. This is partly due to the shift of air-fuel ratio setting from rich, at full load, to slightly lean in the mid-range of engine loads. At low load, the NO_x mass emission trend is reversed and NO_x reaches a minimum at about 10 percent of maximum power, mainly due to mixture enrichment. The sharp increase in NO_x near idle is attributed to the low power output level of the engine. The CO mass emissions appear to be directly related to the air-fuel ratio distribution. The high HC emission at about 30 percent of maximum power may be due to fuel maldistribution effects.

For comparison, test data obtained from that engine at reduced engine speed (1200 rpm) are plotted in broken lines in Figure 3-16. Again, the emissions are referenced to the full load values at 2300 rpm. At 1200 rpm and 45 percent of maximum load, the specific NO_x emissions are about 50 percent lower than at the rated conditions of the engine. However, this improvement is accompanied by a five-percent loss in specific fuel consumption.

Figure 3-17 (Ref. 3-9) shows the effect of engine load, expressed in terms of brake-mean-effective-pressure (BMEP) on the specific mass emissions of a large spark ignition gas engine. As indicated, NO_x decreases rapidly as BMEP is reduced from the base condition while HC increases. As expected, CO is essentially independent of engine load.

3.2.3.5 Other Polutants

In addition to the gaseous pollutants discussed above, gasoline engines emit a large number of other reaction products, some in trace quantities, such as organic acids, high molecular weight olefins,

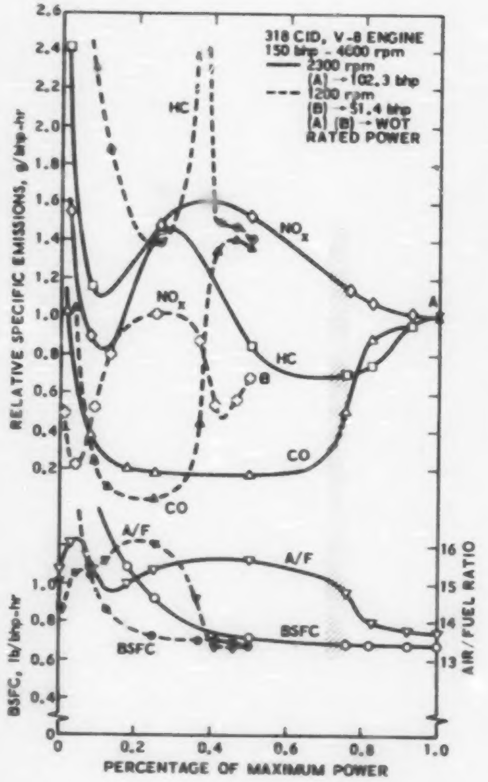


Figure 3-16. Part load emissions of a heavy-duty spark-ignition engine

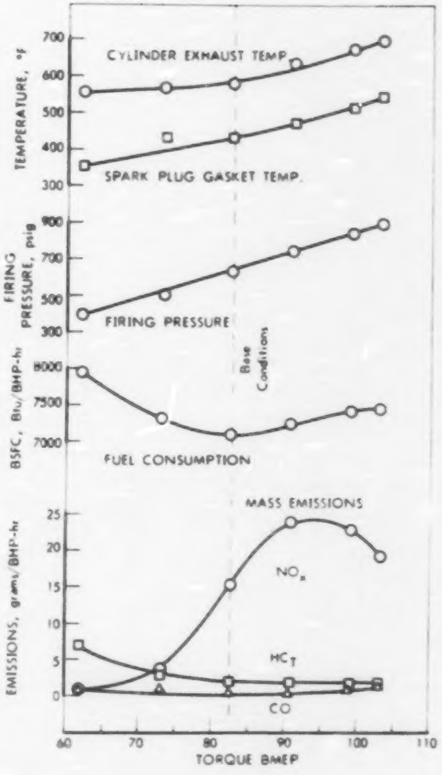


Figure 3-17. Effect of torque on engine performance - large two-stroke spark-gas engine, 300 rpm (Ref. 3-9)

and carbonyl compounds. some of these compounds are believed to be carcinogenic while others contribute to engine exhaust odor. The sulfur contained in the fuel is emitted in the form of sulfur oxides and sulfates. Lead added to gasoline as an anti-knock compound is emitted as lead oxide. Under fuel-rich operating conditions, considerable quantities of carbon particles may be exhausted. In some engines, small amounts of lubricating oil entering the combustion chamber are partially oxidized during the combustion process and are exhausted as a bluish smoke.

In gas engines lead is nonexistent, carbon particles are virtually eliminated, and odor is greatly reduced.

3.2.3.6 Average Emission Values

In the previous sections the emissions at rated conditions and at part loads were listed for several heavy duty gasoline engines and large stationary gas engines. The average emissions for each group are presented in Table 3-16. These values are representative

TABLE 3-16. AVERAGE SPARK-IGNITION ENGINE EMISSIONS AT RATED CONDITIONS

Spark Ignition Engine Type	Specific Mass Emissions g/Bhp-hr			BSFC		A/F
	HC	CO	NO_x	lb/ bhp-hr	Btu/ bhp-hr	
Heavy Duty Gasoline	1.9	23.1	12.2	0.52	–	14.9
Medium-large Gas	4.3	3.2	17.5	–	8000	–
Large Gas	1.1	0.3	22.3	–	7450	22.4

of engines in new or well-maintained condition. The statistical emission data obtained on a large number of automotive engines indicates that the emissions from spark ignition engines increase substantially as the mechanical condition of the engine deteriorates due to wear in extended use or lack of proper maintenance.

3.3 GAS TURBINES

3.3.1 Engine Description

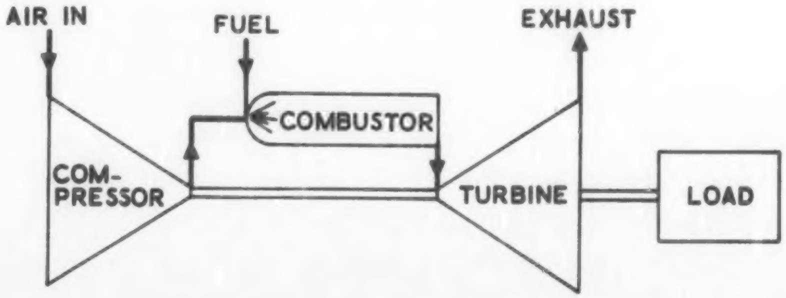
The stationary gas turbine derived its application from aircraft jet engines; and the first gas turbines installed for commercial use were initially derated aircraft gas turbines. Since 1965, gas turbine use for stationary application has greatly expanded as the confidence in their operation grew and as larger units designed specifically for stationary applications became available. In its simplest form, the gas turbine engine consists of a compressor (usually multistage, axial) in which the air is compressed to 100 to 200 psig, followed by a combustor in which fuel (liquid or gaseous) is added for combustion which occurs at overall lean mixtures in the range of fuel-air ratio of 0.01 to 0.02. The combustion gases are then expanded through a turbine or turbines to near atmospheric pressure, converting their energy into power. The design and performance of the combustor is critical not only for the combustion efficiency but also for the level of emissions generated there. This will be discussed in Section 4.3.3. The magnitude and uniformity of the gas temperature at the turbine inlet are also important parameters which, along with the compression ratio, determine the engine efficiency. The efficiency and the power output per pound of air increase with increasing turbine inlet temperature, but a limitation is imposed by the material and stresses in the turbine blades. Typical turbine inlet temperatures for today's stationary gas turbines are of the order of 1800 to 1900°F. The turbine, which can be single or multi-stage, drives the compressor. In a single-shaft engine, excess power is generated by the turbine to drive an electric generator, pump, or any other power absorbing machine. Conversely in two-shaft engines, a second separate turbine is provided downstream of the compressor turbine for power extraction.

3.3.1.1 Simple Cycle

A schematic of a simple cycle gas turbine is presented in Figure 3-18. The typical simple cycle efficiency is 25 to 30 percent. Gas turbines used in today's modern electric utilities typically operate at heat rates (heat input to output power level ratios) of 11,000 to 12,000 Btu/kwh, corresponding to thermal efficiencies of 31 and 28 percent, respectively. However, there are simple cycle gas turbines in operation now which exceed these values, reaching thermal efficiencies of 33 to 34 percent (Ref. 3-72).

3.3.1.2 Regenerative Cycle

The efficiency of simple cycle gas turbines can be upgraded by further extraction of heat from the exhaust gases which normally leave the turbine at 800 to 1100°F. One of the ways of doing so is with a regenerative cycle as shown in Figure 3-19 (Ref. 3-73).



THERMAL EFFICIENCY, 25 TO 30%

Figure 3-18. Simple cycle gas turbine (Ref. 3-73)

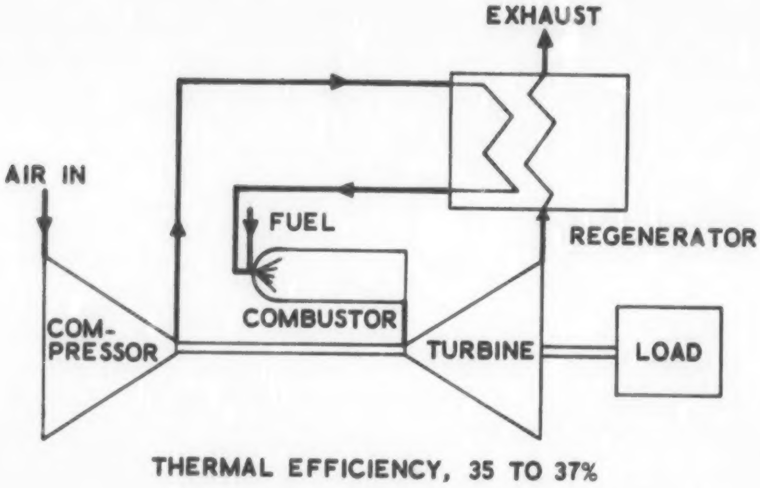


Figure 3-19. Regenerative cycle gas turbine (Ref. 3-73)

Air leaving the compressor is fed to the combustor via a heat exchanger in which some of the heat of the exhaust gases leaving the turbine is transferred to the pressurized air so that at fixed turbine inlet temperature less fuel has to be added. This improves the thermal efficiency to about 34 to 38 percent, corresponding to heat rates of 10,000 to 9,000 Btu/kwh.

3.3.1.3 Combined Cycle

Another way of utilizing the exhaust waste heat is in a combined cycle shown schematically in Figure 3-20 (Ref. 3-73). In this case, the exhaust gases are used to produce steam in a steam generator which can then be used to drive a steam turbine to produce additional power. The usual power generation split is: 60 percent gas turbine; 40 percent steam turbine. In some cases, for higher quality steam, an afterheating device is added by burning additional fuel in the exhaust (fired versus unfired cycle). The combined cycle has the

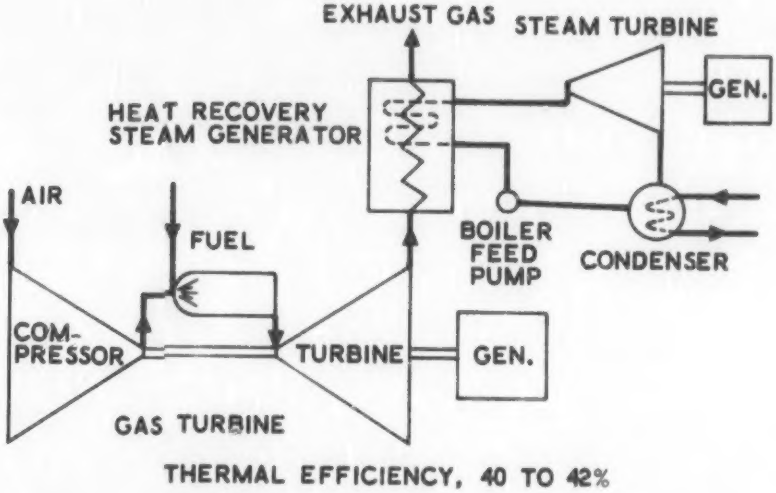

EXHAUST GAS
STEAM TURBINE
GEN.
HEAT RECOVERY STEAM GENERATOR
AIR
FUEL
BOILER FEED PUMP
CONDENSER
COM-PRESSOR
TURBINE
GEN.
GAS TURBINE
THERMAL EFFICIENCY, 40 TO 42%

Figure 3-20. Combined cycle gas turbine and steam generator (STAG) system (Ref. 3-73)

highest thermal efficiency, with some of the existing units operating at 40 to 42 percent, corresponding to 8530 to 8100 Btu/kwh heat rates.

3.3.1.4 Future Design Trends

All types of gas turbine cycles discussed above are in operation. Of course, the simple cycle has the longer operating experience behind it while the combined cycles started operation only recently. Before discussing the various applications, a few comments are presented on the future design philosophy of stationary gas turbines.

The future stationary gas turbines will be subjected to a number of controlling parameters such as: a) rules regarding exhaust emissions, b) requirement to operate on residual oils and possibly coal, c) increase in plant size for intermediate load electric power generation, and d) higher performance to offset rising fuel costs. The problem of emissions will be discussed in detail in Section 3.3.3.3; suffice it to say here that there are opposing trends, with low NO_x dictating lower combustion temperature while high efficiency implies higher temperatures and compression ratios. Obviously, a compromise between these two conflicting requirements has to be found and it will consist mainly in improvements in the combustor, in the premixing of the fuel with air, and in a reduction in the reaction temperature and residence time.

The projected improvements in thermal efficiency of simple and regenerative cycle gas turbines as affected by higher turbine inlet temperature and compression ratio are shown in Figures 3-21 and 3-22. Turbine inlet temperatures of 2400°F and higher are predicted with improved materials and intercooled compressor air bleed for turbine cooling, resulting in a simple cycle engine efficiency of up to 40 percent. For comparison, modern steam-electric plants have design heat rates of 9500 Btu/kwh (36-percent efficiency) and future projections are up to 8500 Btu/hr (40-percent efficiency) for large plants (1000 MW).

The combined cycle efficiency will increase as a result of a simple cycle efficiency increase and higher exhaust temperature level, and in the next 10 years heat rates as low as 7000 Btu/kwh, corresponding to an efficiency of approximately 50 percent are predicted (Ref. 3-74). This is not unexpected since Curtiss-Wright presently offers combined cycle MOD POD 75 units generating 99 MW at a thermal efficiency of 44.8 percent (7650 Btu/kwh) (Ref. 3-74).

With the gas turbine assuming not only peak loads but also intermediate loads in electric power generation, its size will be increasing. At present, the largest gas turbine power plant consisting of two identical units has a capacity of 480 MW (Ref. 3-75). In future plants such as the combined cycle plant discussed in Reference 3-76 the power level will be raised to 1000 MW. Costs estimates for a plant consisting of four units of either simple cycle or regenerative cycle configuration are presented in Ref. 3-77. The projected cost varies

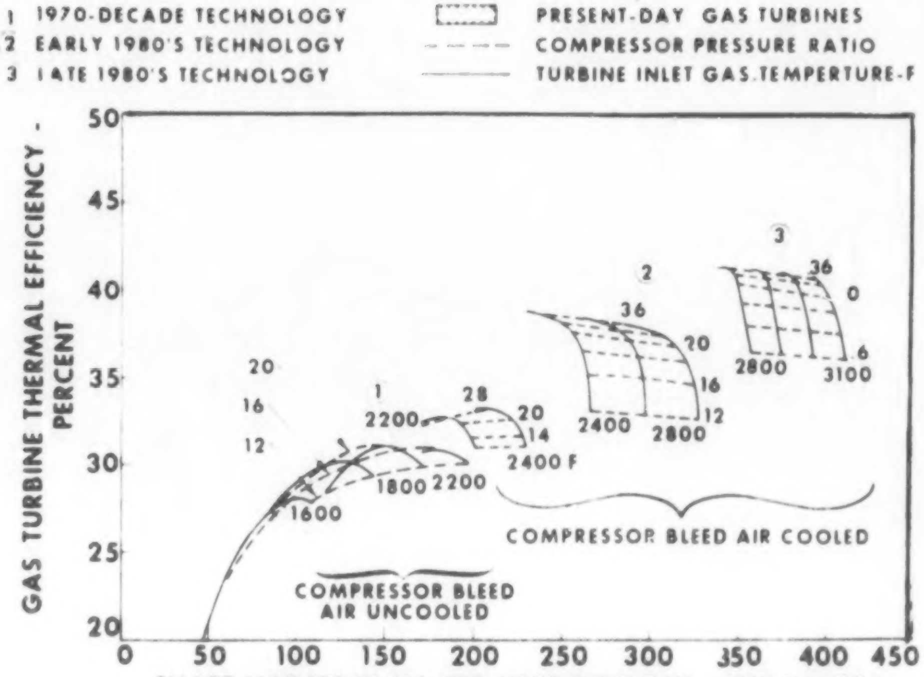


Figure 3-21. Simple-cycle gas turbine performance (Ref. 3-77)

from \$66.21/kW for simple cycle units to \$92.69/kW for regenerative units (1970 dollars).

With the looming shortage of light fuel distillates and natural gas, it is to be expected that there will be a growing requirement for gas turbines capable of burning heavy fuels or fuels derived directly from coal.

A study of a combined cycle plant using coal (named "Cogas") was made in Ref. 3-75. Gasification of coal would generate low Btu gas (150 to 200 Btu/scf) which could be produced at an investment cost (over that of the cost of coal) of 22¢/10^6 Btu. Active development plans for building such a plant are underway, under sponsorship

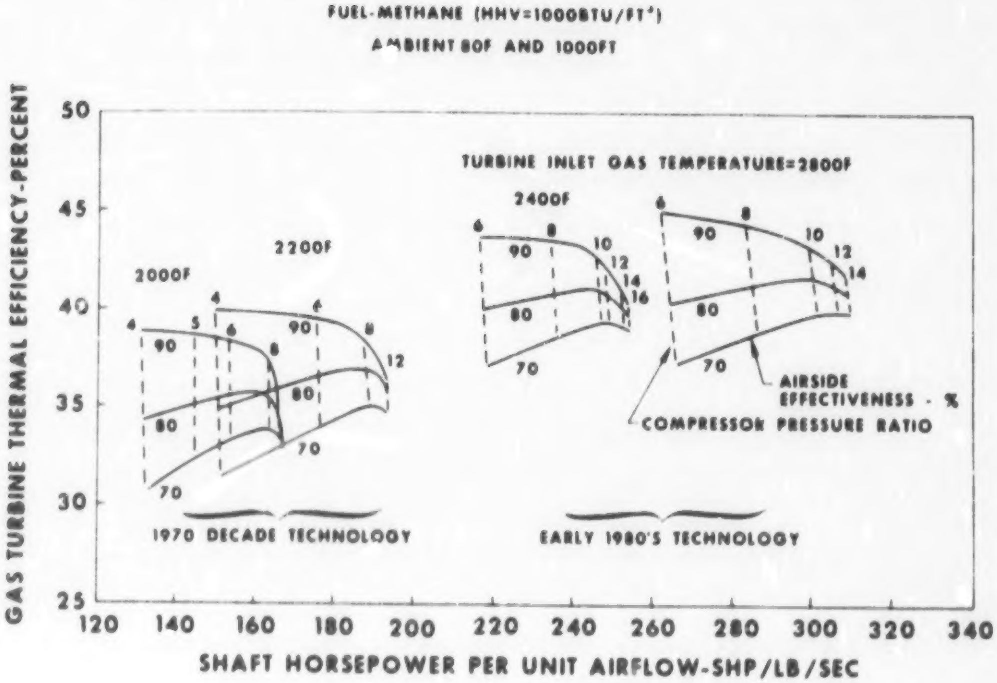


Figure 3-22. Regenerative-cycle gas turbine performance (Ref. 3-77)

of the Office for Coal Research with an objective of having a 135-MW pilot plant in operation in mid-1976 (Ref. 3-78). The plant would operate with a turbine inlet temperature of 2200°F and a cycle efficiency (unfired boiler) of 47 percent.

Another development which should be useful in permitting emission-controlled combustion of heavy and residual oils or coal is the closed Brayton cycle turbine under development by the Garrett Corporation under sponsorship of the U.S. Department of Commerce. The present program is aimed at maritime applications and will culminate in a closed air cycle turbine developing power in the range of 40,000 to 80,000 hp (Ref. 3-79). In this design the air is heated in an external combustor thus permitting great flexibility in fuel selection and emission control.

3.3.2 Applications

The main applications of stationary gas turbines include electric power generation for utilities and industrial use, pipeline service, and repowering. These are discussed in the following sections.

3.3.2.1 Electric Power Generation

An electric utility's demand grows daily from a nighttime low to higher levels in the day, with peaking at certain hours. Figure 3-23 shows the three basic types of electric power generation: base load, which is continuous operation; intermediate load consisting, typically, of 2000 to 6000 hours/year operation; and peaking load at about 1000 to 2000 hours/year (Ref. 3-80). The peaking units have a reserve load capability rating which is 10 to 15 percent above the nominal rating. In addition, there are standby emergency units which typically operate less than 100 hours/year.

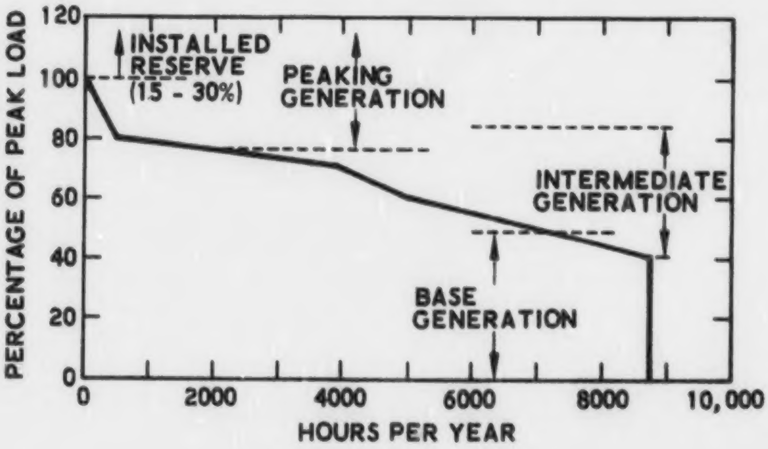


Figure 3-23. Electric power generation schedule (Ref. 3-80)

Gas turbines have been generally accepted by the utilities in the last several years as peak load shavers, but it was only in the last 2 to 3 years that they began to be installed for intermediate loads. There are several reasons to explain the growing popularity of gas turbines and Ref. 3-81 quotes about 20 of them. The most important are: low initial cost; short delivery time; non-dependence on cooling water for simple cycle and regenerative cycle machines; small plant area; flexibility in operation on various fuels; and high thermal efficiency of the regenerative and combined cycle.

The investment cost of a simple cycle gas turbine is below \$100/kw (for more details see Section 5.3) which is less than half of the investment cost of a steam plant. The operating cost will depend on the percent of gas turbine utilization and on fuel cost.

The delivery time for a large gas turbine plant is approximately two years compared to four years for a steam turbine (Ref. 3-77). This, combined with a modular build-up in which most of the assembly of the critical components is done in the factory for subsequent shipment to the site in the form of modules, reduces the investment cost of gas turbines.

The size of the gas turbine plant is considerably smaller than that of a corresponding steam plant and Ref. 3-77 quotes 0.006 acres/MW for a regenerative cycle plant which amounts to about one-tenth of the requirement for a conventional steam plant. Table 3-17 presents a comparison of the power density of different type plants (Ref. 3-82).

Most of the gas turbines in the above applications operate with dual fuel systems (No. 2 GT fuel and natural gas) and the power turbine speed, whether in a single- or two-shaft engine, is geared to 3600 rpm generators (60 Hz current). The typical Btu/kwh data given before correspond to fuel-to-air ratios of approximately

TABLE 3-17. POWER DENSITY COMPARISON (Ref. 3-82)

Plant Type	Power Density (kw/ft^3)(fuel to power conversion equipment only)
Reciprocating engines	0.01
Fossil - steam turbine	0.07
Nuclear - steam turbine	0.23
Combined cycle (STAG)	0.84
Simple cycle gas turbine	3.50

0.018 to 0.022 for simple and combined cycles operating at nominal rating, and 0.012 to 0.016 for regenerative configurations.

Another feature of the gas turbine is its quick start capability. In a combined cycle, the gas turbine can be at full load in about 20 minutes with the steam turbine following in another 20 minutes. Ref. 3-75 reports 10 minutes minimum to 30 minutes normal start to full load for a 240 MW gas turbine. For comparison a conventional steam-electric plant requires about four hours.

3.3.2.2 Pipeline Service

Gas turbines are used in pipeline service to drive either compressors (for gas) or pumps (for oil). The power range is between 1000 to 12,000 bhp and the average usage load factor is high (90 percent). Because of the high load factor and the rising cost of fuel, high thermal efficiency is important and regenerative cycles are now frequently used.

3.3.2.3 Re-powering

Gas turbines are used for re-powering when old, less efficient steam plants are being retired or when there is a need for power output increase. Ref. 3-83 lists some cases for re-powering including: (1) replacement of old boilers by a new set heated with gas turbine waste heat, (2) replacement of steam turbines by a gas turbine for an existing generator drive, and (3) gas turbines added to produce power or to drive forced draft fans of existing boilers. Detailed examination of some cases has shown that repowering can not only add power but also improve cycle efficiency of old steam plants from values less than 25 percent to better than 35 percent.

3.3.2.4 Installed and Projected Power

3.3.2.4.1 Electrical

As shown in Figure 3-24 (Ref. 3-81) the increase in the installed gas turbine power has proceeded very rapidly for the reasons discussed in Section 3.3.2.1. The trend toward larger units is reported in Ref. 3-81 and this trend will continue as the gas turbine assumes intermediate and, in some cases, base load functions. High efficiency cycles will also be in a growing demand and Ref. 3-74 reports that as of December 1973 at least 15 combined and 15 regenerative cycle units were on order by various utilities.

The total installed gas turbine power capacity in 1970 was 16,500 MW compared to all generating power sources of 331,000 MW; that is, approximately 5 percent (Ref. 3-81). The projected growth of turbine power is shown in Figure 3-24, forecasting approximately 69,000 MW in 1980 (total U. S. capacity in 1980 is estimated to be 600,000 MW). The total U. S. electrical power in 1986 was estimated by the Federal Power Commission at 760,000 MW (Ref. 3-84) which would increase the gas turbine contribution to approximately 9 percent. The average rate of gas turbine growth between 1973 and 1980 is thus estimated at approximately 5300 MW/year, which compares with the average growth of 6000 to 7000 MW/year in the last two years. The next few years growth may be tempered somewhat with the growing shortage of distillates and

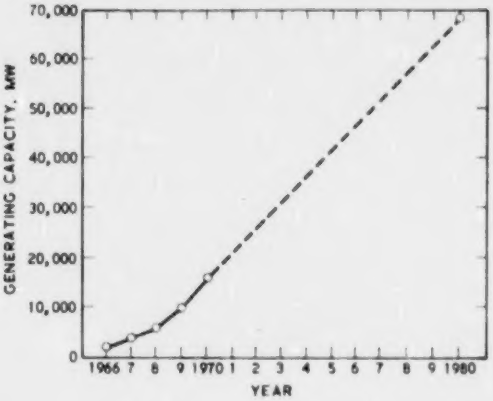


Figure 3-24. Projected gas turbine generating capacity, megawatts, in United States 1966-1980 (Ref. 3-81)

natural gas for use in gas turbines; but this will probably trigger more effort on the part of the engine manufacturers for adaptation of gas turbines to heavy distillates, oil crudes and residuals, and also to the coal gasification products (i.e., Cogas plant). If this occurs, uninterrupted growth of gas turbine power could be maintained.

3.3.2.4.2 Pipelines

In pipeline service the gas turbines are used for oil pumping, drilling, gas compression, and oilwell cementing. The horsepower installed in the years 1958-1970 is shown in Table 3-18.

The greater portion of the installed power is for gas compression and oil pumping. The recent trend is towards larger power units, as shown in Figure 3-25. In 1971 the installed gas compressor and oil pumping horsepower was 22.2×10^6 hp (Ref. 3-85) of which 3.5×10^6 hp or 15.9 percent was generated by gas turbines and

TABLE 3-18. GAS TURBINE POWER FOR PIPELINE USE, 1958-1970 [a] (Ref. 3-81)

Horsepower Range	Number of Gas Turbines	Total Installed hp x 10^3
1000 - 4999	1260	1908
5000 - 9999	470	3416
1000 and larger	260	3422
TOTAL	1990	8746

[a] A summary table listing the installed horsepower of all stationary engines is presented in Appendix A.

12.2×10^6 or 55.2 percent by reciprocating engines with the remainder consisting mainly of electric power. A rather steady growth in the rate of pipeline gas turbine power can be expected in the future. The development of the Alaskan pipeline will require additional power as will the offshore drilling. For these applications, the gas turbine provides the lowest cost and weight per horsepower unit. For instance, the average cost of installed gas turbines for a compressor station is \$250/hp versus about \$400/hp for reciprocating engines (Ref. 3-81).

The operation of pipeline units is mainly continuous and the gas turbines perform well at the high load factor showing smaller initial, repair, and operating cost than reciprocating engines (Ref. 3-81).

3.3.2.4.3 Re-powering

In the era of growing fuel cost, the incentive for the installation of gas turbines is increased thermal efficiency (reduced fuel cost/kwhr) accomplished by combined cycle systems utilizing gas

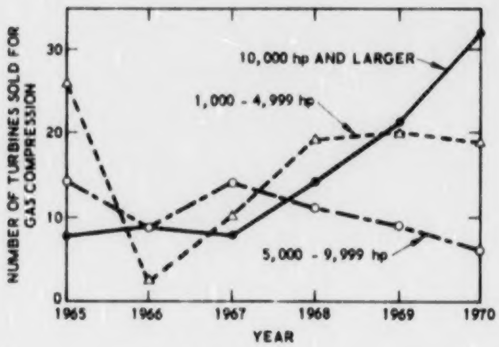


Figure 3-25. Trends in size of turbines sold for gas compression service (Ref. 3-81)

turbine waste exhaust heat for steam production. Ref. 3-86 quotes approximately 800 MW re-powering with gas turbines by various utilities in 1972.

3.3.2.4.4 Gas Turbine Manufacture and Operation

The life of a stationary gas turbine is usually estimated at 20 years. However, this figure may vary somewhat depending upon the duty cycle utilized. The maintenance record of gas turbines is good and Ref. 3-87 quotes data from the operational experience of a combined cycle, showing typical inspection intervals of 7500 hours for the combustor, 15,000 hours for the hot gas section, and 30,000 hours for major inspections. The steam turbine generator is inspected annually and overhauled every 10 years.

The manufacturers of gas turbines and important engine characteristics are listed in Ref. 3-81 for both domestic and foreign manufacturers. Table 3-19 summarizes most of the domestic manufacturers of stationary gas turbines.

TABLE 3-19. STATIONARY TURBINES – U.S. MANUFACTURERS

Manufacturer and Type	Power Range (unit rating)	Application (domestic only)	Cycle	Fuel[a]
Allison-General Motors				
Model 404	300 - 350 hp	Pipeline	Simple	NG, DO
Model 501-K	4000 - 5000 hp	Pipeline Electrical	Simple	NG, DO, DF
Model 912F	18,000 - 22,000 ~20 MW	Pipeline Electrical	Simple	NG, DO, DF
Avco-Lycoming				
TF 12	1150 hp	Pipeline	Simple	NG, DO
TF 14	1400 hp	Pipeline	Simple	NG, DO
TF 25	2250 hp	Pipeline	Simple	NG, DO
TF 35	2800 hp	Pipeline	Simple	NG, DO
TF 40	3350 hp	Pipeline	Simple	NG, DO
Curtiss-Wright				
Mod-Pod 20/25	24,000 - 31,000 hp	Pipeline	Simple	NG, DO, DF
	20 - 25 MW	Electrical	Simple	NG, DO, DF
Mod-Pod 30	35,000 hp	Pipeline	Simple	NG, DO, DF
	27 MW	Electrical	Simple	NG, DO, DF
Mod-Pod 40/50	40 - 50 MW	Electrical	Simple	NG, DO, DF
Mod-Pod 60	65 MW	Electrical	Simple	NG, DO, DF
Mod-Pod 75	70 MW	Electrical	Simple	NG, DO, DF
TEC-150	150 MW	Electrical	Combined	NG, DO, DF
TEC-350	350 MW	Electrical	Combined	NG, DO, DF
General Electric				
Model 1000	4000 - 5000 hp	Pipeline	Regenerative	NG
Model 3000	8000 - 15,000 hp	Pipeline	Simple and regenerative	NG
	5 - 15 MW	Electrical	Simple, compound, regenerative	NG, RO, DF
Model 5000	12,000 - 24,000 hp	Pipeline and process	Simple and regenerative	NG, DF
	11 - 26 MW	Electrical	Simple	NG, DO, DF
Model 7000	45 - 83 MW	Electrical	Simple, regenerative, combined (STAG)	NG, DO, DF
Model 9000	80 - 91 MW	Not for domestic application	Combined (STAG)	NG, DO, DF

[a] NG - natural gas
RO - residual oil
DO - distillate oil
DF - dual fuel (oil/gas)
RG - refinery gas

TABLE 3-19 - continued

Manufacturer and Type	Power Range (unit rating)	Application (domestic only)	Cycle	Fuel[a]
Solar				
Spartan	225 kW	Electrical	Simple	NG, DO, DF
Saturn	1100 hp	Pipeline	Simple	NG, DO, DF
	750 - 800 kW	Electrical	Simple	NG, DO, DF
Centaur	3000 - 3300 hp	Pipeline	Simple	NG, DO, DF
	2250 kW	Electrical	Simple, regenerative	NG, DO, DF
Turbo Power and Marine Systems				
FT4A	20,000 - 30,000 hp	Pipeline	Simple	NG, DO, DF
	20 - 23 MW	Electrical	Simple	NG, DO, DF
FT4C	35,000 - 42,000 hp	Pipeline	Simple	NG, DO, DF
	25 - 32 MW	Electrical	Simple	NG, DO, DF
TP4-2 Twin-Pac	45 - 55 MW	Electrical	Simple	NG, DO, DF
TP4-4 Twin-Pac	90 - 110 MW	Electrical	Simple	NG, DO, DF
TP4-9 Twin-Pac	180 - 220 MW	Electrical	Simple	NG, DO, DF
Turbo-Steam Pac	75 - 125 MW	Electrical	Combined	DO, RO
Westinghouse				
Models: W-21, W-31, 41, 52	1800 - 5300 hp	Pipeline	Simple, regenerative	NG, RG
Models: W-62, W-72, 81, 82, 92	5000 - 9500 hp	Pipeline	Simple, regenerative	NG
Models W-101, W-121, W-171	7 - 15 MW	Electrical	Simple, or with heat recovery	NG, DO, DF
Model W-191	14.5 - 17 MW	Electrical	Simple, or with heat recovery	NG, DO
Model W-251	18.5 - 33 MW	Electrical	Simple	NG, DO, DF
Model W-501	37 - 65 MW	Electrical	Simple	NG, DO, DF
Model - Pace 20, 30	17.5 - 30 MW	Electrical	Combined	NG, DO
Model - Pace 260	240 - 260 MW	Electrical	Combined	NG, DO, DF

[a] NG - natural gas
RO - residual oil
DO - distillate oil
DF - dual fuel (oil/gas)
RG - refinery gas

3.3.3 Emissions

Before reviewing gas turbine emissions, some of the existing or proposed emission standards are discussed to provide a yardstick for the emission problem assessment.

3.3.3.1 Federal and State Standards

The Federal Emission Standards for stationary power plants are defined by the Federal Register, Vol. 36, No. 247, Part II of 23 December 1971. These standards apply to new stationary sources defined as "Fossil-Fuel Fired Steam Generators" and, therefore, are not applicable to gas turbines. However, since standards for gas turbines are now in the process of preparation, it is of interest to review these standards since there will probably be some similarity between these and the standards under consideration for gas turbines. Table 3-20 presents a partial list of currently applicable emission standards. (Ref. 3-88.)

Table 3-21 presents conversion factors for the various emission units computed for typical gas turbine fuel-air ratios.

It should be noted that the emissions should always be referenced to a standard day condition and correlations for the conversion from a current day to a standard day must be provided. Tests carried out indicate the dependence of the emission level on temperature and the following correction formulae were suggested in Ref. 3-89:

$$CO_{\text{std day}} = CO_{\text{meas}} \times \theta^{6}$$

$$HC_{\text{std day}} = HC_{\text{meas}} \times \theta^{13}$$

$$NO_{x\ \text{std day}} = NO_{x\ \text{meas}} \times \frac{1}{\theta^{4}}$$

TABLE 3-20. EXAMPLE OF EMISSION STANDARDS (MAXIMUM ALLOWABLE)

Emission / Agency	NO_x	CO	SO_2	Smoke	Particulates	Remarks
Federal Register	0.2 lb/10^6 Btu - gas 0.3 lb/10^6 Btu - liquid 0.7 lb/10^6 Btu - solid		0.8 lb/10^6 Btu - liquid 1.2 lb/10^6 Btu - solid	20% opacity	0.1 lb/10^6 Btu	Applicable to plants with minimum of 250 × 10^6 Btu
City of Los Angeles Rule 67	140 lb/hr		200 lb/hr		10 lb/hr	
San Diego County Rule 50				20% opacity Ringelmann #1		
Rule 52					0.1 grains/SCF	After January 1974
Rule 62			0.5% per weight of sulfur in fuel			
Rule 68	125 ppm (3% O_2) - gas 225 ppm (3% O_2) - liquid					Applicable to plants with 50 × 10^6 Btu/hr minimum
Proposed EPA (Ref. 3-90)	55 ppm (15% O_2) - gas 75 ppm (15% O_2) - liquid	90 ppm (15% O_2) or 215 ppm (15% O_2) (plants < 50 × 10^6 Btu/hr)	145 ppm (15% O_2) or 0.8 lb/10^6 Btu	20 % opacity for fired combined cycle 10% opacity for simple cycle and unfired combined cycle		

The ppm values are defined at specified percent O_2 in the exhaust. The actual percent O_2 in gas turbine exhaust at base load is approximately 15%. The conversion to reference percent O_2 is $ppm_{actual} \times [21 - \%O_{2(ref)}]/[21 - \%O_{2(actual)}] = ppm_{ref}$. Thus: ppm (3% O_2) = 3 × ppm (15% O_2).

TABLE 3-21. GAS TURBINE EMISSION UNITS
(11, 500 Btu/kwh; 18,500 Btu/lb fuel)

Units / Emission	ppm (F/A = 0.0166) (measured)	ppm (15% O_2) (reference)	lb/1000 lb fuel	lb/10^6 Btu	g/hphr
NO_2	100	112	9.5	0.51	1.9
CO	100	112	5.8	0.31	1.2
SO_2	100	112	13.2	0.71	2.7
HC (as C)	100	112	2.5	0.13	0.5

where:

$$\theta = \frac{T_{\text{inlet meas}}}{T_{\text{inlet std day}}} \; ; \; T_{\text{inlet std day}} = 519°R.$$

To correct for humidity the following correlation has been proposed by Marchionna (Ref. 3-89).

$$\frac{NO_{x \text{ meas}}}{NO_{x \text{ zero humid}}} = e^{-19 \cdot H}$$

where H = lb of water/lb of dry air. Based on this equation a 17.3 percent reduction in NO_x is realized for each 1 percent of water vapor increase in the ambient air.

3. 3. 3. 2 Gaseous Emissions

3. 3. 3. 2. 1 Overview

Before examining in more detail the stationary gas turbine emissions, it would be useful to get a perspective of them with regard to the total emission problem in the U.S. In particular, the comparison will be made for NO_x which is the predominant pollutant for stationary sources and probably most difficult to control.

In 1968, stationary sources contributed approximately 60 percent to the total of 16 × 10^6 tons of NO_x (calculated as NO_2) emitted in the U.S. (Ref. 3-91). These numbers are in reasonable agreement with data published in Ref. 3-92 (10 × 10^6 tons of NO_x for stationary sources and 20.6 × 10^6 ton total NO_x emission). Since the NO_x emissions from gas turbines are substantially lower than those of stationary reciprocating engines and boilers, the total gas turbine emissions in that period were less than 1 percent of all stationary sources (Ref. 3-85 and 3-92).

The comparison of the growth in the total generated power and in gas turbine power until 1980, which is needed for an estimate of future gas turbine emissions, is shown in Table 3-22.

With regard to HC and CO, these emission species are insignificant for gas turbines compared to other stationary sources (Ref. 3-85). Moreover, the HC and CO contribution of all stationary

TABLE 3-22. TOTAL AND GAS TURBINE STATIONARY POWER

	Electrical			
	1971	1972	1975	1980
Total Installed Power, MW	367,396	399,606	516,700[a]	657,000[a]
Gas Turbine Power, MW	21,774	27,918	42,000[b]	69,000[b]
	Other Stationary (pipeline, industrial)			
Gas Turbine Power, hp × 10^{-3}	6,313	6,628[c]	7,672[c]	9,792[c]

[a] These data based on fossil and nuclear power plants already committed for this period (Ref. 3-84). Further power increase in the 1975-80 period probable with additional plant order (FPC estimates electric generating power at 760,000 MW in 1980).

[b] From Ref. 3-81.

[c] 5 percent/year growth assumed from 1971 value.

combustors compared with all other emission sources is rather small, amounting to only about 2 percent in 1968 (Refs. 3-93 and 3-94).

The emission of particulates from hydrocarbon fuels (mainly coal) in all stationary sources was 47 percent of the total particulate emission in the U.S. (Ref. 3-95). Again, the gas turbine contribution is insignificant because of operation with clean fuels. These aspects are further discussed in Section 3.3.3.4.

The emission of SO_x from stationary sources is appreciable because of the high sulfur content in some of the fuels used, particularly coal and oil crudes. In 1968 the contribution of all stationary sources amounted to 75 percent of the total SO_x emission in the U.S. On the basis of Btu burned, stationary gas turbines would contribute about 3 percent of the total SO_x emitted by stationary units. In reality, their SO_x emission will be much lower (about one percent) because of the low sulfur fuel (natural gas, No. 2 GT oil) used in most gas turbines. The most economic control of SO_x is related to fuel - desulfurization, as discussed in Section 4.3.5. Thus, it appears that while a control on stationary gas turbine emissions is certainly beneficial in preventing high local NO_x or SO_x concentrations, the nationwide impact of such controls will not be significant.

3.3.3.2.2 Emissions

The gas turbine combustors may be divided into the primary or combustion zone and the post-flame or secondary zone. The post-flame zone can be subdivided into a thermal soaking zone and the final dilution zone in which the excess air lowers the gas temperature to a value acceptable for use in the turbine rotating assembly (Ref. 3-96). The principal pollutant species emitted from gas turbines are NO_x, followed by CO, HC, and SO_2. These are discussed in the following sections.

3.3.3.2.2.1 NO_x

The formation of NO_x in gas turbines proceeds by three known mechanisms, characterized as "hot air," "prompt," and "organic" (Ref. 3-97). The relative importance of these three mechanisms on NO_x is affected by the fuel composition, fuel injector, flame temperature, fuel-air ratio, size and shape of the combustor, and gas flow patterns.

The "hot air" NO is produced in the hottest regions of the combustor in accordance with the Zeldovich reactions:

$$N_2 + O \rightleftarrows NO + N$$

$$N + O_2 \rightleftarrows NO + O$$

The studies on the formation of NO led to three important conclusions (Ref. 3-98): The NO formation rate is exponentially dependent on flame temperature while pressure and oxygen concentrations have a secondary effect; NO is frozen at kinetically limited concentrations when the combustion gases are quenched; the decomposition rate of once-formed NO is so slow that it is not significant in a combustion system.

It appears, therefore, that a key to low NO_x is the achievement of a uniformly low flame temperature in the primary combustor coupled with a low residence time. This is illustrated in Figures 3-26 and 3-27. The flame temperature in the primary zone can be lowered by operation with lean fuel-air mixtures with equivalence ratios in the primary zone of 0.8 or less. The experimental work with advanced lean combustors in which the kerosene fuel was premixed with air and prevaporized (to reduce the residence time), indicated that a substantial decrease in NO_x (with an attendent reduction in CO) was indeed achieved (Ref. 3-99).

As mentioned above, the rate of formation of NO is kinetically limited and, for instance, at 4025° F flame temperature it would take over one second to reach an equilibrium of 6000 ppm of NO_x

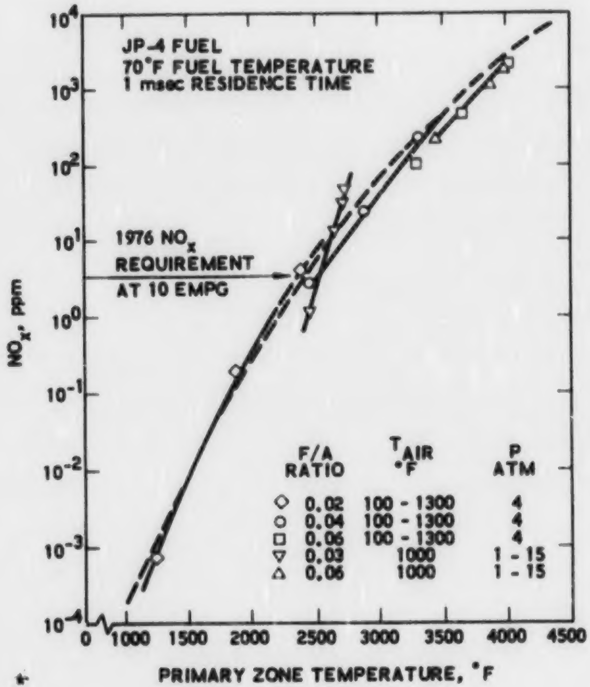


Figure 3-26. Theoretical effect of primary zone temperatures on NO$_x$ emissions (Ref. 3-98)

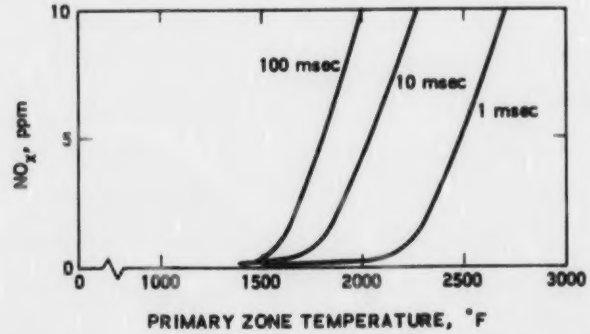


Figure 3-27. Combined effect - residence time and primary zone temperature (Ref. 3-98)

(Ref. 3-100). Limiting the residence time and temperature will reduce the NO appreciably, as shown in Figure 3-27.

The second mechanism of "prompt" NO formation is a rapid but very short duration reaction involving highly reactive free carbon or hydrocarbon radicals (Ref. 3-96). The mechanism, which is not yet completely understood, must occur during combustion and not in the hot combustion products, and measurements in laboratory flames and estimates in combustion have shown "prompt" NO to be about one-third of the amount produced by the "hot air" mechanism (Ref. 3-97).

The third mechanism of "organic" NO formation is from the chemically-bound nitrogen in fuel molecules. Because the nitrogen-carbon bond energy in the fuel molecules is so much lower than that of molecular nitrogen, much of the fuel nitrogen becomes oxidized. These reactions may proceed at lower temperatures than needed for the two former mechanisms, and organic NO is less responsive to the abatement techniques now in use and more to the fuel composition (Ref. 3-97).

More data on fuel bound nitrogen are given in Section 3.3.3.6. However the fraction of nitrogen converted to NO is not firmly established. Ref. 3-101 quotes for stationary boilers 30 percent nitrogen conversion for fuel-oil, and 50 percent for coal. Ref. 3-102 gives the conversion factor as a function of total nitrogen content of fuel oil. Thus, for instance, fuel with 0.1 percent nitrogen should contribute approximately 30 ppm of NO_x (based on 15 percent O_2 in the exhaust) in a typical gas turbine.

A substantial amount of emission data, particularly NO_x, was obtained by the powerplant manufacturers and users. A few of the data are presented below, showing the effects of fuel type (for example, fuel oil versus natural gas) and operating cycle (simple, regenerative, control/overloaded).

Based on the mechanisms of NO_x formation, as well as analytical work by various researchers, it appears that NO_x should be lower at part-load operation and when using gaseous fuels. Gaseous fuel operation approaches the ideal condition obtained in premixed, vaporized, and well-stirred combustors. This is illustrated in Figure 3-28 presenting measurements from a 20 to 22 MW simple cycle gas turbine. Over the whole load range the maximum NO_x (no abatement measures) was around 230 ppm with No. 2 fuel oil and about half of that for gas fuel (see Figure 3-29). In these tests the nitric oxide was measured by the PDS (phenoldisulfonic acid) method (ASTM-D1608-60). Similar results are shown in Ref. 3-103 from another manufacturer's 35 MW gas turbine fired with No. 2 fuel oil. The measurements of NO_x were determined in this case by wet chemical means with the PDS and modified Saltzmann methods, and by on-line gas analyzers. Figure 3-30 shows NO_x emissions for two different size gas turbines, 26 and 66 MW. Figure 3-31 was prepared from various data obtained by Aerospace and illustrates: a) increased NO_x level for increased plant size because of increased residence time, b) 30 to 50 percent lower emission with fuel gas than fuel oil, and c) emission reduction by 20 to 30 percent obtained with new combustor designs. Advanced combustors are further discussed in Section 4.3.3.3.

Of interest are the emission data collected in Ref. 3-104 on gas turbines (and spark ignition engines) driving pipeline compressors. The data are for nine gas turbines varying in horsepower from 1000 to 20,000. NO_x was measured by the PDS and chemiluminescence methods, CO by NDIR (dispersive infrared), and HC (expressed as pure carbon) by flame ionization. The data of Ref. 3-104 are summarized in Figure 3-32 in terms of ppm and g/hp-hr, the latter being more characteristic for mechanical drive engines. The data indicate that the emissions of gas turbines are, on the average, an order of magnitude lower than those from uncontrolled spark ignition engines.

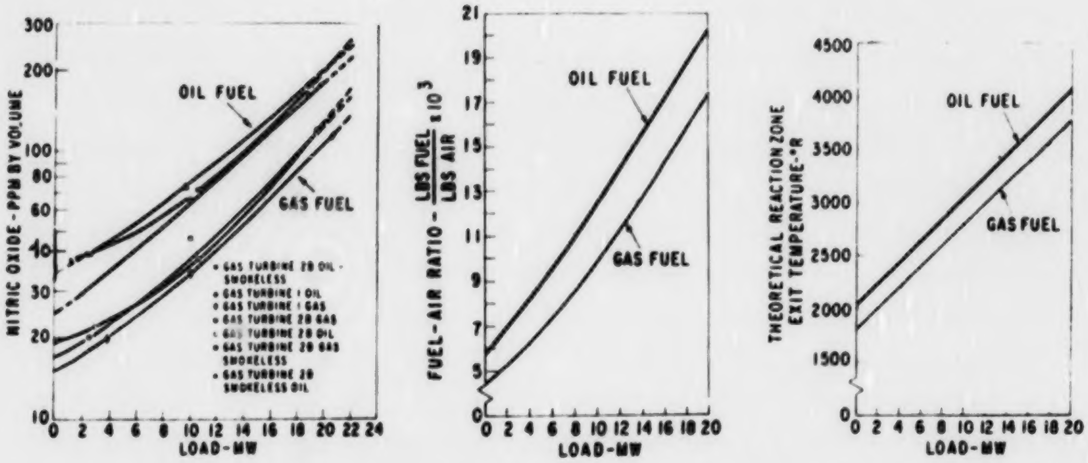


Figure 3-28. Operating characteristics at part and full-load of a 22 MW gas turbine (Ref. 3-96)

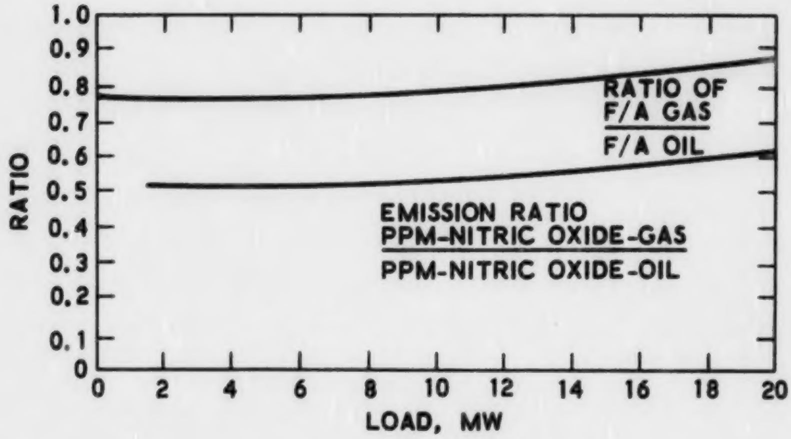


Figure 3-29. Nitric oxide emission ratio and fuel-air ratio versus load (Ref. 3-96)

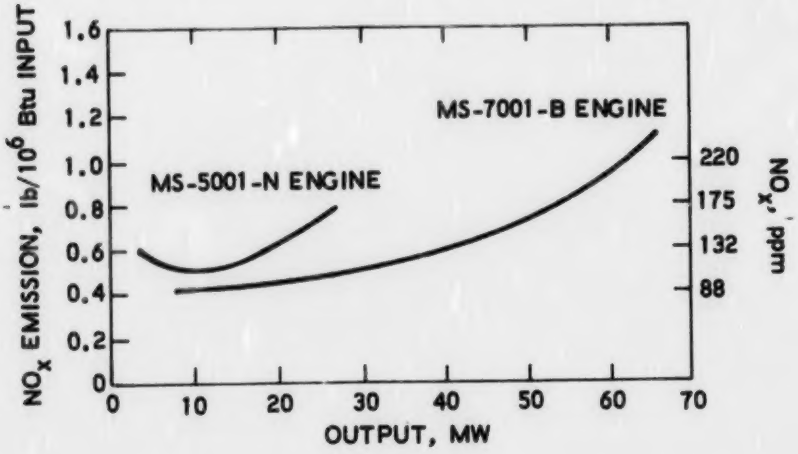


Figure 3-30. NO$_x$ emissions – simple cycle oil-fired (Ref. 3-73)

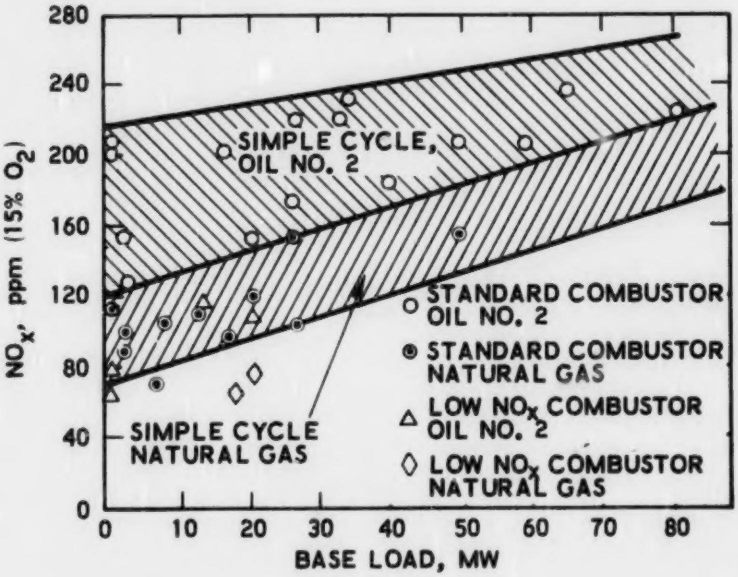


Figure 3-31. Typical gas turbine NO$_x$ emission at base load

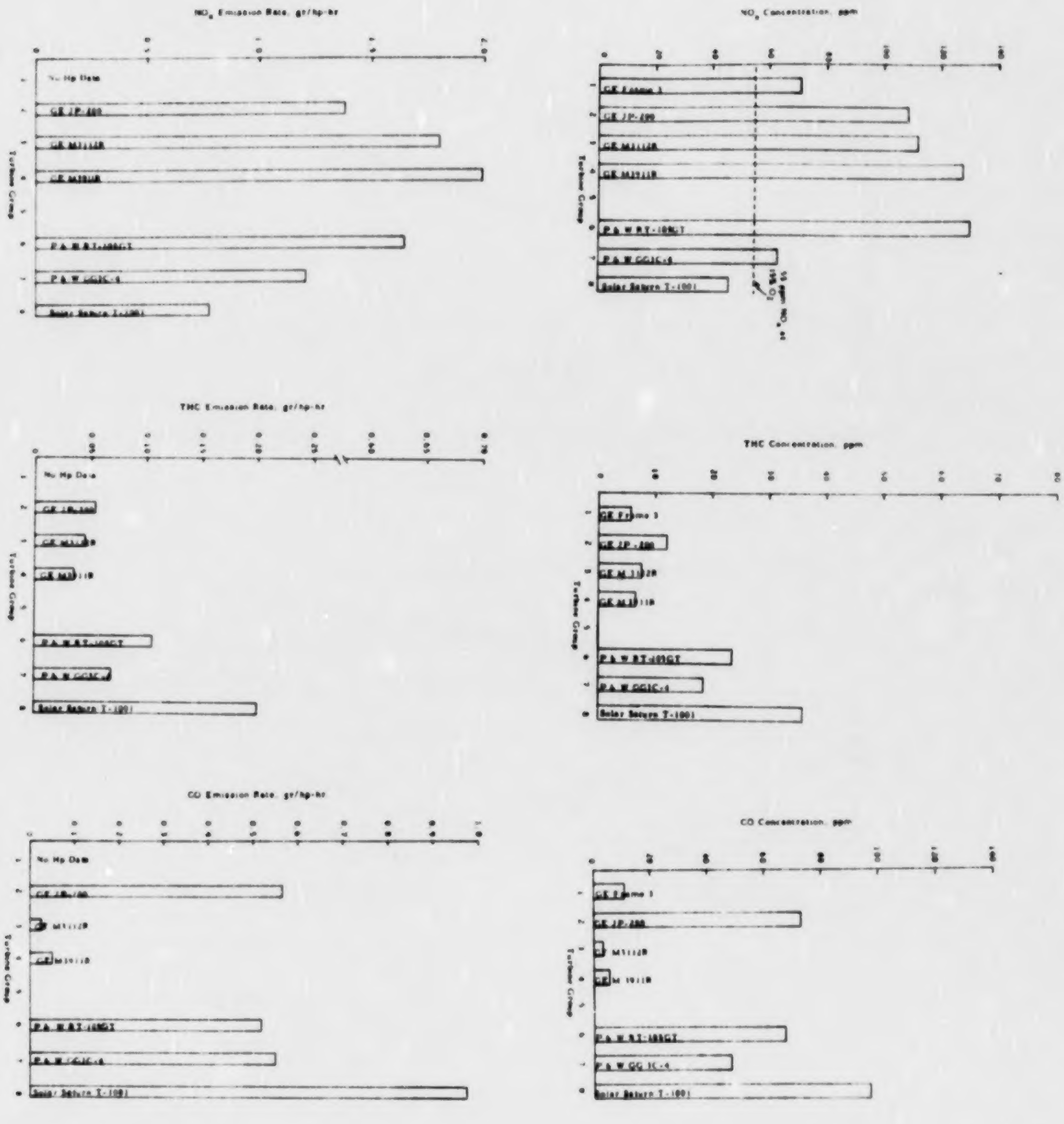


Figure 3-32. Emission data of pipeline gas turbines — natural gas (Ref. 3-104)

The NO_x emissions of a regenerative cycle gas turbine as compared to a simple cycle engine are subject to two opposing effects. The temperature of the air entering the combustor is higher than in a simple cycle, which would imply higher NO_x formation rates. On the other hand, with a fixed turbine inlet temperature, less fuel is required in a regenerative cycle (which accounts for higher efficiency) and the primary combustion zone will run leaner with lower flame temperature which would imply a lower rate of NO_x formation. Thus, the regenerative and simple cycle engines should have similar NO_x emissions (Ref. 3-103 and 3-105). However, this conclusion is not general and is a function of the combustor design, air flow pattern, fuel type, etc. For instance, Figure 3-33 shows higher NO_x emissions for a natural gas-fired regenerative gas turbine than for simple cycle engines.

The combined cycle with unfired exhaust will have similar NO_x emission characteristics as that of a simple cycle because of the previously discussed low rate of NO decomposition. The exhaust-fired combined cycle burns additional fuel but since the exhaust gases have a reduced oxygen content, the flame temperature is lower even though the initial temperature is higher. Since the rate of formation of NO is highly

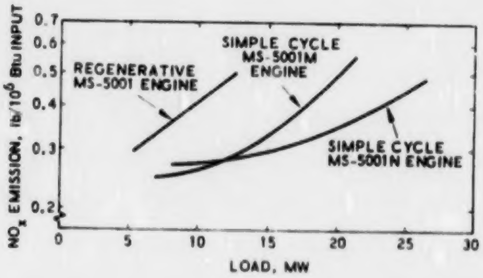


Figure 3-33. NO_x emissions, natural gas-fired, iso-conditions (Ref. 3-73)

temperature-dependent, less NO per megawatt is produced than in the main combustor of the gas turbine (Ref. 3-82).

3.3.3.2.2.2 CO

Carbon monoxide is formed by incomplete combustion (oxidation) of carbon and/or dissociation of CO_2. After forming, it can be subsequently reduced by further oxidation to form carbon dioxide. Similarly to the reaction rate between oxygen and nitrogen, the oxidation rate of carbon monoxide is highly temperature- and time-dependent. At lower temperatures, the oxidation rate of CO is slower, and if the residence time is short more CO will be left in the combustion products.

Thus, in general, conditions favoring low NO_x will tend to increase the CO content and here lies the difficulty in the design of low emission combustors. Fortunately, most stationary gas turbines operate at or near base load and the combination of high temperatures and high excess air (typically 200 to 300 percent) results in relatively low CO emission.

In general, concentrations of CO are higher at idle and low power settings because of inadequate mixing, low temperature, and fuel quenching on the walls. At higher power levels, some CO may be formed by dissociation of CO_2 at high temperatures in the primary zone and be "frozen in" at levels higher than local equilibrium because of the quenching effect from the dilution air (Ref. 3-106).

Figure 3-34 shows the CO emissions as a function of load. The water injection effect shown in the figure will be discussed in Section 4.3.4. The measurements of CO were obtained by an on-line analyzer which responds to the concentration of gas by an electrochemical reaction through a membrane sensor.

Figure 3-35 presents CO emission data collected from industry for fuel oil and natural gas-fired gas turbines. One would expect a lower CO emission with gas because of better mixing, but this doesn't seem to be the case, perhaps because of a lower primary zone

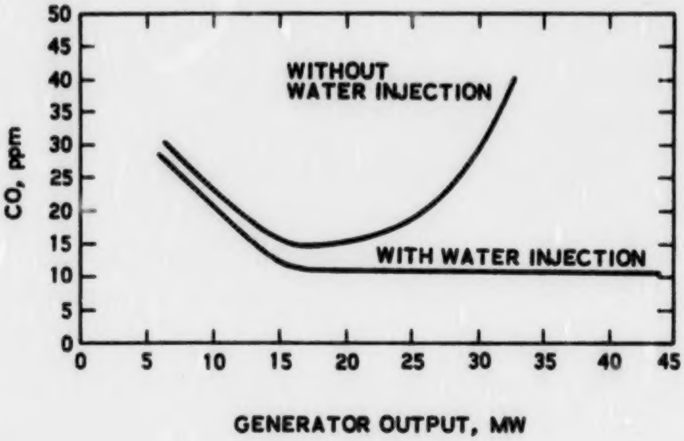


Figure 3-34. CO versus load, W-251 engine (Ref. 3-103)

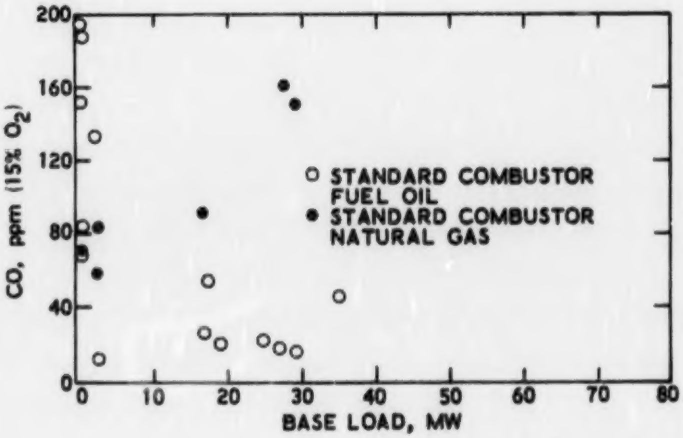


Figure 3-35. Emissions for various gas turbine powerplants

temperature and the lower flammability limit obtained with gas as compared to fuel oil (see Figure 3-28). There is an increase of CO in small combustors because of the attendent shorter residence times available for further oxidation.

The conflicting set of combustor requirements for low NO_x concurrent with low CO is illustrated in Figure 3-36. For example, high CO at idle is accompanied by low NO_x. The arrow in this figure indicates the path for low emission combustor development. A similar $CO\text{-}NO_x$ relationship is illustrated in Figure 3-37 (Ref. 3-107).

The emission of CO is expected to be lower with a regenerative cycle because of higher combustor inlet temperatures (Ref. 3-108).

3.3.3.2.2.3 HC and Aldehydes

The hydrocarbons are formed in a gas turbine combustor in a manner similar to that of carbon monoxide. HC is the result of incomplete combustion caused by poor mixing, inadequate fuel distribution and atomization, and wall quenching. Its formation is a function of temperature, and both CO and HC will tend to behave in a similar manner.

Figure 3-38 shows HC data from various gas turbines. Similarly to CO, there is no apparent relationship between HC and fuel type (fuel oil or gas) but some increase in HC is noted for small engines.

The data on aldehyde emissions are rather limited. In general they are a small proportion of the total HC emissions which are very low in most gas turbines. Aldehydes are formed by partial oxidation of hydrocarbon fuels combined with the formation of free radicals which results in a subsequent formation of the aldehyde group (CHO). Typically, for fuel oil the aldehyde emissions amount to approximately one-third of the total HC emissions (based on a colorimetric method of measurement) (Ref. 3-109). Like HC the level of aldehydes declines with increasing load.

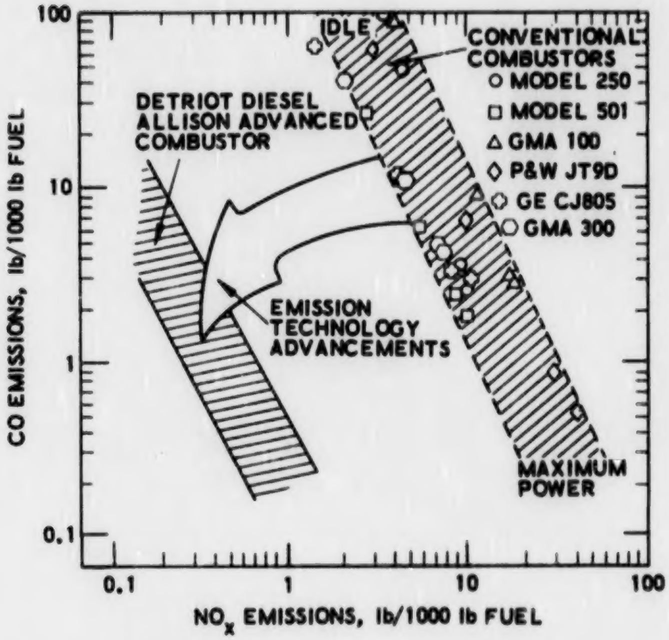


Figure 3-36. CO versus NO$_x$ emission performance of conventional gas turbine engine combustors (Ref. 3-107)

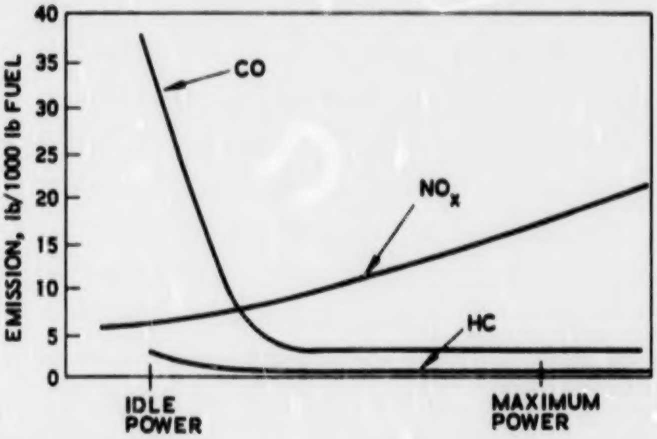


Figure 3-37. GMA 100 gas generator emissions (Ref. 3-107)

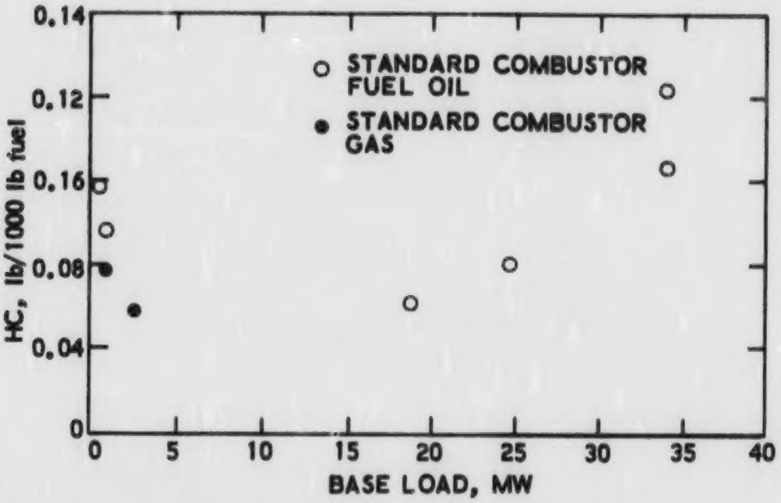


Figure 3-38. HC emissions for various gas turbine powerplants

3.3.3.2.2.4 SO_x

The oxides of sulfur emitted from gas turbines are produced during the oxidation process of the fuel which is the sulfur donor. Thus, the amount of sulfur oxide emissions are a direct function of the fuel sulfur content and nothing can yet be done in a practical sense to reduce it. Thus, the Federal Emission Standard for SO_2 (0.8 lb/10^6 Btu for liquid fuels) led to an upper limit of 0.73 percent sulfur in a liquid fuel. This can be compared to presently established sulfur limits for hydrocarbon fuels, shown in Table 3-23.

The combustion of sulfur results mainly in sulfur dioxide, although 3 to 4 percent are further oxidized to sulfur trioxide which readily dissolves in water to form sulfuric acid (Ref. 3-105).

Figure 3-39 shows SO_2 measurements as a function of load. The change in ppm with load is caused by varying the fuel-air ratio. In this case SO_2 was measured by grab sampling and by an

TABLE 3-23. FUEL SULFUR CONTENT IN PERCENT (Ref. 3-73)

Fuel	Minimum	Average	Maximum	ASTM Maximum
Natural Gas	-	-	0.010	-
ASTM No. 1 GT (kerosene)	0.018	0.039	0.098	0.5
ASTM No. 2 GT (fuel oil)	0.070	0.270	0.550	0.5
ASTM No. 3 GT (heavy oil distillates)	0.200	-	1.920	-

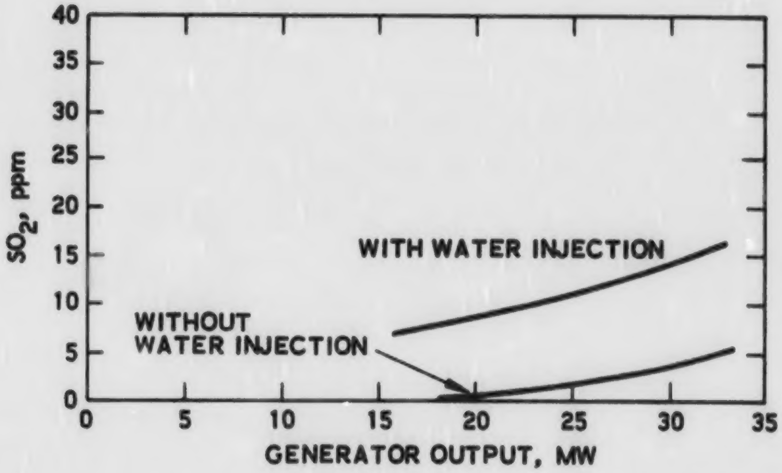


Figure 3-39. SO_2 versus load, engine W-251 (Ref. 3-103)

on-line analyzer, using membrane-type polarographic sensors. The small difference in SO_x (10 ppm) with and without water injection is probably due to different collection methods, i.e., "dry" versus wet.

The emission of SO_2 could become a problem in the future because of the continuing increase in total power and fuel consumption and a gradual change from no-sulfur natural gas to high sulfur crudes and residuals (Ref. 3-109 and 3-110).

Since the SO_2 emission is a function of fuel consumption, all types of plants are equally affected. The only plant-dependent parameter would be its thermal efficiency and, of course, higher efficiency would result in less SO_2 emission at a given power output.

3.3.3.2.2.5 Typical Stationary Gas Turbine Emissions

From the information presented in Section 3.3.3.2, typical emission data for medium-size (30 MW) gas turbines with state-of-the-art combustors are summarized as follows:

NO_x - (#2 oil)	150-220 ppm (15% O_2)
- (natural gas)	90-140 ppm (15% O_2)
CO - (oil or gas)	5-100 ppm (15% O_2)
HC (as C) - (oil or gas)	1-20 ppm (15% O_2)

For smaller units, NO_x decreases while CO and HC may show an increase. The opposite is true for larger combustors. Techniques to reduce the emission levels (NO_x in particular) to acceptable values are discussed in Section 4.3.3.

3.3.3.3 Smoke, Particulates and Odor

3.3.3.3.1 Smoke

Gas turbine exhaust smoke which consists of small carbon particles is a result of incomplete combustion in locally fuel-rich zones (Ref. 3-111). The carbon particles forming the visible smoke are 0.01 to 1.0 microns in size.

Several factors in the gas turbine combustor contribute to the generation of smoke. These are: coarse liquid fuel atomization, fuel-rich pockets in the primary combustor, low flame temperatures, insufficient fuel-air mixing and fuel composition. In general, heavy fuels are greater smoke producers while natural gas is almost smoke-free. In gas turbines, the most favorable conditions for smoke formation occur at idling when the atomization of liquid fuel is poor and the temperatures are low. Some smoke may also be generated at full load when the fuel-air ratios are higher.

Various methods for smoke measurements have been developed and some are described in Refs. 3-112 and 3-113. The Bacharach smoke number is obtained from the ASTM-D-2156 method which specifies filtration of a standard volume of exhaust through a Whatman paper filter (Ref. 3-103). The resulting smoke spot stain is evaluated photometrically from zero (100-percent reflectance from an unstained filter patch) to nine (10-percent reflectance from a heavily stained patch). The Ringelmann method consists of visual comparison of smoke by trained observers against four charts with numbers from one to four, corresponding to increasing smoke density. The von Brand smoke number is a measure of the reflected light from a smoke sample on filter paper taken under controlled conditions (Ref. 3-73). The reading of 100 corresponds to no smoke and zero to optically black conditions.

Typically the limit of smoke visibility in large gas turbine exhaust stacks is at Bacharach No. 5 (Ref. 3-114), corresponding to about 20 percent opacity which is defined as a limit in current Federal Regulations. Figure 3-40 shows typical smoke emissions from a 33 MW gas turbine, and Figure 3-41 shows smoke in von Brand numbers as a function of particulates. The reduction of smoke with water injection resulted from some of the combustor contaminants being dissolved in water. Both units shown in Figure 3-41 operate at base load at von Brand numbers of about 90 which is below the visibility threshold.

The present day gas turbines can, in general, meet the smoke limits imposed by various States (Ref. 3-88) without difficulties, but future operation on heavier fuels may require more development work.

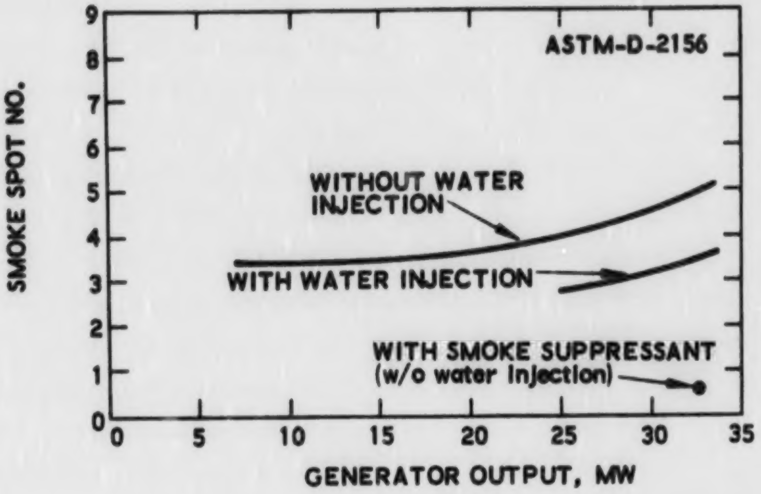


Figure 3-40. Smoke versus load, engine W-251 (Ref. 3-103)

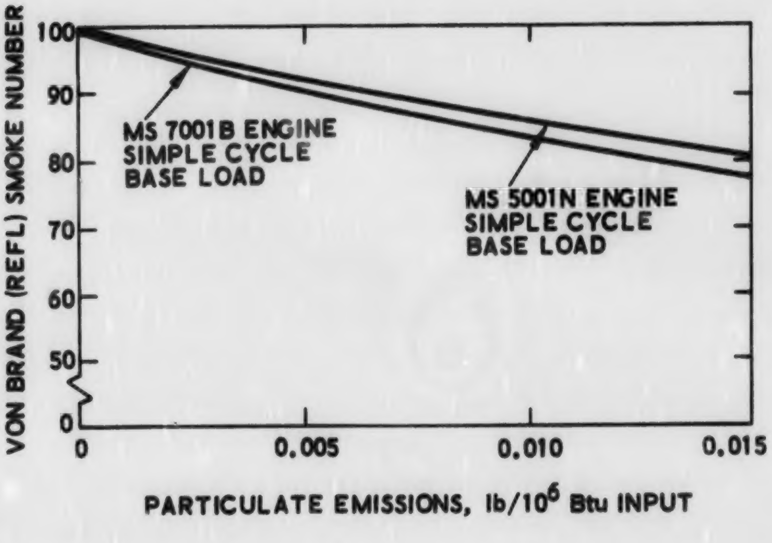


Figure 3-41. Calculated particulate matter emission rate resulting from black smoke particles versus von Brand (reflectance) smoke number (Ref. 3-73)

3.3.3.3.2 Particulates

With regard to particulate matters, gas turbines produce relatively low amounts because of the type of fuel burned and the high efficiency of the combustion process (Ref. 3-115). Light distillate fuel oils burn relatively cleanly and contain very small amounts of ash and trace materials, and gaseous fuels are even better. Carbonaceous and sulfur-related particulate species account for the majority of the particulate matter obtained with distillate fuels. The former can be controlled by the combustor design and the latter by using low sulfur fuel. When burning heavy, residual fuels, the particulate emission attributed to fuel ash and fuel contaminants can be appreciable (Ref. 3-115). The typical nature of particulate matter and its origin is shown in Table 3-24.

Various methods for particulate emission measurement exist and a few of those (EPA, ASTM, LAAPCD) are described in Ref. 3-98. They can be divided into "wet" methods where a sample of exhaust is discharged through a filter and wet impingers which are subsequently heated to remove the water, leaving behind the residues; and into "dry" methods where the particulates are collected on dry, heated

TABLE 3-24. BREAKDOWN OF PARTICULATE MATTER (Ref. 3-115)

Particulate Matter	Source
Smoke (carbonaceous)	Fuel cracking
Ash and trace metals	Fuel content and fuel additives
H_2SO_4, $2H_2O$, XSO_4[a]	Fuel sulfur
Organics	Unburned fuel
Ambient noncombustibles	Turbine air inlet
Erosion products	Material from gas flow path surface

[a] Considered particulate matter in only a few localities.

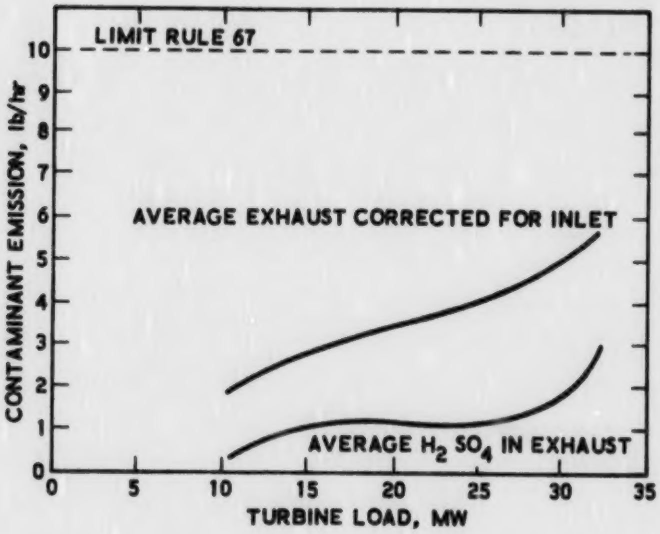


Figure 3-42. W-251 engine combustion contaminants (dry filter method) versus load (Ref. 3-103)

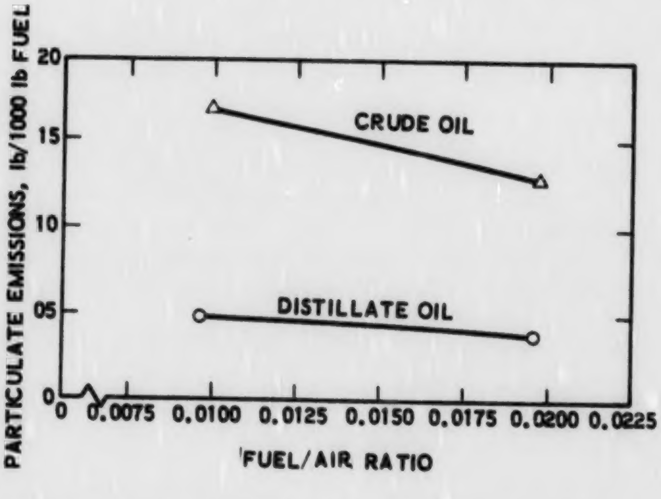


Figure 3-43. Particulate matter emission when burning crude and distillate oil fuel (Ref. 3-103)

filters. According to Ref. 3-103, the "wet" methods may show an excessive particulate content because of combined water, manufactured sulfates and sulfated metal compounds. This was shown by the use of both "wet" and "dry" methods of measurement.

Figure 3-42 shows the particulate emission (or combustion contaminants) as a function of load using the "dry" filter methods of measurement. It can be seen that the limit of Rule 67 can be comfortably met.

Figure 3-43 shows particulate emissions for distillate and crude oil as a function of fuel-air ratio. It is of interest to note that while a 33 MW gas turbine consuming approximately 30,000 lb/hr of distillate oil would meet the Rule 67 contaminant limit (10 lb/hr), it would be much above the limit with crude oil. This confirms a comment made earlier on heavy fuel operation.

3.3.3.3.3 Odor

The odor from gas turbine exhaust is noticeable in high traffic density airports and some aspects of it have been investigated by NASA (Ref. 3-116). It is not a problem in stationary gas turbines but a few comments on the NASA investigations are appropriate.

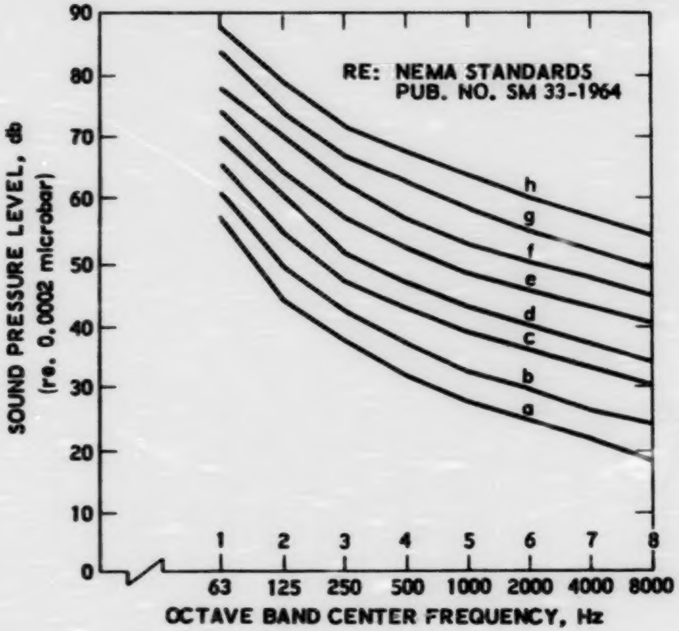
Exhaust odor intensity from gas turbines was evaluated by a human panel and graded progressively from the threshold level to ranking No. 3. The odor is created by fuel aromatics and by oxygenated compounds such as alcohols, aldehydes, and ketones (Ref. 3-116). Test data indicate that the odor increases with the concentration of oxygenates as well as with combustion inefficiency. There was also dependence on fuels in that high aromatic JP-5 had the highest odor rating while natural gas had the lowest. The quality of liquid fuel atomization had an effect on odor by affecting combustion efficiency, and air-assisted fuel injection was especially effective in reducing the odor intensity.

3.3.3.4 Noise

Noise is a pollutant of some importance in stationary gas turbine installations. The advantage of the high-power density of a gas turbine power plant and the flexibility of its installation, which is independent of water supply for simple and regenerative cycles, also makes possible its location at populated load centers. With this siting advantage comes the necessity of quiet operation (Ref. 3-82). What other power generation plants gain in sound reduction by remoteness, gas turbines must provide with silencing systems.

The sources of the gas turbine noises include: the inlet; the rotating machinery (compressor and turbine); and the exhaust. The noise generated by these three main contributors is attenuated by means of a suitable silencing system to meet the local sound criteria. In addition to the gas turbine generated noise, there is noise from the power absorbing machinery such as electric generators and gas compressors which also require silencing. Most of the engine manufacturers were guided by the NEMA (National Electrical Manufacturers Association) noise standards for industrial and residential centers. These standards are shown in Figure 3-44. The standards applied to a given installation will depend on the plant location and usually the initial cost of a plant includes noise silencing to a given NEMA level. New standards are being prepared by the American National Standards Institute (ANSI).

Figure 3-45 presents typical noise emissions of a stationary gas turbine compared against NEMA standards for industrial and nearby residential installations. The noise emitted falls between the industrial and residential limits and achievement of lower values than those shown requires some additional form of silencing. The range from the source to the listener is important in noise attenuation and if the range is large compared to the length of the source, the noise is reduced by 6 db every time the distance is doubled (Ref. 3-117).



TYPE OF AREA	PEAKING DAYTIME ONLY	PEAKING NIGHTTIME ONLY	CONTINUOUS DAYTIME ONLY	CONTINUOUS DAY & NIGHT
HEAVY INDUSTRY	h	f	g	e
URBAN -- NEARBY INDUSTRY	g	e	f	d
URBAN -- RESIDENTIAL	f	d	e	c
SURBURBAN -- RESIDENTIAL	e	c	d	b
VERY QUIET -- SURBURBAN OR RURAL RESIDENTIAL	d	b	c	a

Figure 3-44. NEMA noise standards for industrial and residential centers (Ref. 3-118)

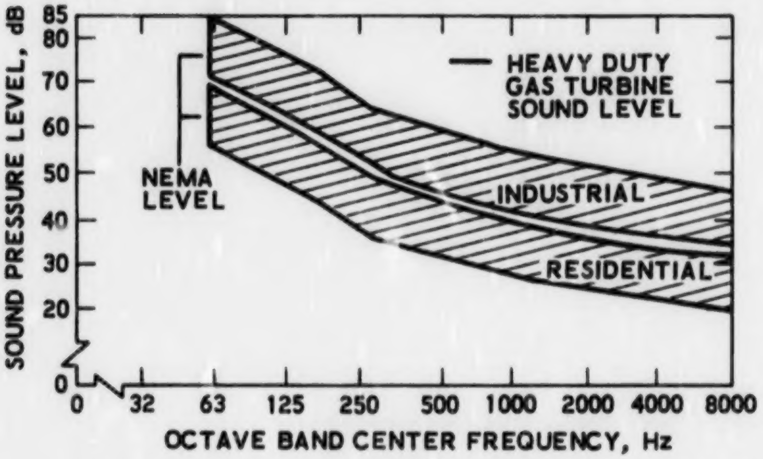


Figure 3-45. Sound level performance of heavy-duty gas turbines compared to NEMA sound levels (Ref. 3-82)

The "acceptable" versus "unacceptable" noise intensity depends on many factors such as noise level (decibels), frequency, duration, time of occurrence (day versus night), background noise level, etc. A survey of community response was made in Ref. 3-118 and compared against the proposed sound treatment levels by a gas turbine manufacturer.

The combined cycle and regenerative gas turbines provide their own noise attenuation. In a combined cycle, the steam generation chamber attenuates noise either by the use of absorptive materials or by the reverberant room effect, while the regenerator absorbs noise by creating vortices in the flow passage through which some of the acoustic energy is converted, and by partial scattering and reflection. This attenuation may become as high as 15 db at high frequencies (Ref. 3-119).

Various approaches to noise legislation are described in Ref. 3-120 and it appears that the technology is available to allow stationary gas turbine compliance with most of the existing noise standards.

3.3.3.5 Emissions Versus Gas Turbine Operating Conditions

The stationary gas turbines operate, in most cases, at nearly constant load level around the base or peak load design points. Consequently, the emission control is believed to be somewhat easier than for automotive gas turbines which have to operate over the whole load spectrum.

The conflicting set of conditions leading to either low CO and HC or low NO_x are reflected in the gas turbine emission characteristics showing high values of CO and HC at idle and peak loads, with much lower values at design loads. The opposite is true for NO_x (Figures 3-36 and 3-37). With high CO and HC at idle, the appearance of smoke becomes more pronounced, and in a number of operational units air blast atomization was added to improve the quality of fuel atomization and to reduce the smoke to an acceptable level.

The trend in future gas turbines to improve cycle efficiency and reduce fuel consumption will be in the direction of higher compression ratios, higher turbine inlet temperature and in the continuing development of regenerative and combined cycles. While this trend has a beneficial effect on CO and HC, it will make the control of NO_x more critical because of the exponential relationship between the rate of NO_x formation and flame temperature. However as shown analytically in Ref. 3-121 and confirmed experimentally in Ref. 3-99, the use of fuel-air premixing combined with well-stirred combustion and short residence times are very potent means in controlling NO_x.

The regenerative cycle gas turbines operate at lower compression ratio than simple cycles because of the compressed air temperature increase in the regenerator. It appears (Ref. 3-99) that at a given turbine inlet temperature the emission of NO_x may be similar to or lower than that of simple cycle engines.

Combined cycle engines with unfired exhaust have emission characteristics similar to those of simple cycles. With fired exhaust, additional emissions will be generated. However per MW output they should be lower than those of the main combustor.

3.3.3.6 Fuel Effects

The fuels used most frequently in stationary gas turbines include natural gas, kerosene, or naphtha (No. 1 GT) and fuel oil or diesel fuel (No. 2 GT). Future use will include heavy distillates or crude oil (No. 3 GT), residual oils (No. 4 GT), and products of coal gasification.

Natural gas is over 94-percent methane without any metallic or sulfur contaminants. Kerosene is a low carbon-hydrogen ratio fuel with near zero contaminants, low viscosity, and little variability in physical properties. Fuel oil has higher viscosity than kerosene (Saybolt Seconds Universal 34 to 50 at 80°F), a moderate carbon-hydrogen ratio with near zero contaminants and little variability in physical properties. Heavy distillates or crude oils are characterized by high carbon-hydrogen ratio, higher viscosity (Saybolt Seconds Universal 180 to 2000 at 80°F) and contaminants varying from zero to about five ppm. There is also a great variability in physical properties. The residual oils have still higher viscosity (of the order of 10,000 to 20,000 SSU), high carbon-hydrogen ratio, high contaminant level, and wide variability in physical properties (Ref. 3-122). In many instances, the high viscosity of Nos. 3 and 4 GT oils will require preheating of the fuel to lower its viscosity to the 50 to 800 SSU level for good atomization (Ref. 3-123). The product of coal gasification might be low Btu gas (180 to 300 Btu/scf) consisting mainly of CO, H_2, CH_4, CO_2, and about 50 percent N_2. Contaminants and sulfur might be removed in the gasification process.

The current interest in heavy fuels and coal is stimulated by the growing shortage of natural gas and light distillates. This is illustrated in Figure 3-46. With the predicted rise and doubling of the gas turbine fuel demand in the 1973-1979 time period, there will be a reduction in the use of natural gas and a rapid increase in the use of crude and residual oils. The coal gas is not included in this figure since it will enter the picture in the post-1979 period. However, the predictions shown in Figure 3-46 should be viewed with caution. The rapidly growing cost of light distillates may influence the refineries to reduce the amount of residuals and new refineries now being built may not yield more than 2 to 3 percent of the ingoing crude in the form of residuals (Ref. 3-122). With low yield of residuals and a growing demand for it in steam plants and gas turbines, the price differential

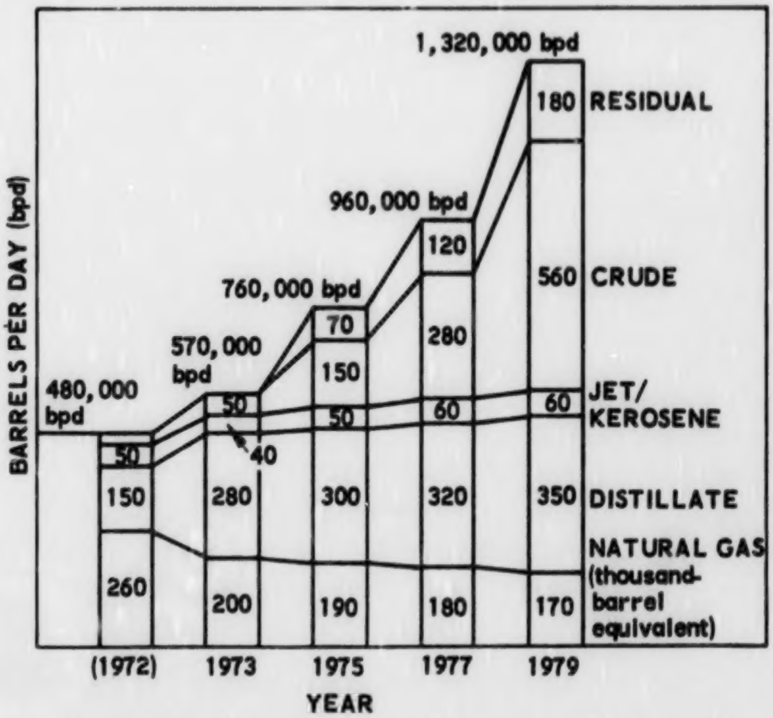


Figure 3-46. Electric utility gas turbine fuel demand (Ref. 3-109)

between No. 3 GT and No. 4 GT fuels may be narrowed considerably. On the other hand, there is a need for pretreatment of the fuel with residual oils. The residuals will contain almost all impurities of the original crude in the form of trace elements such as sodium, potassium, vanadium, and lead (Ref. 3-124), as well as sulfur.

Vanadium and sodium have been shown to cause turbine bucket and nozzle corrosion and a limit of 0.5 ppm has been suggested (Ref. 3-74). Vanadium and sodium limits of 2 ppm are proposed in Ref. 3-123. Whichever limit is valid, it is lower than the contaminants present in the residual oils requiring pretreatment of the fuel. At present this approach consists of washing out the alkali metal salts with water and then separating the water by centrifuging or by electrostatic means (Ref. 3-125). Vanadium can be inhibited by addition of magnesium in a weight ratio of three parts of magnesium to one part of vanadium (Ref. 3-125). The washing and inhibiting techniques reportedly eliminate turbine corrosion as a major problem, but it increases the total amount of combustion contaminants (Ref. 3-105). These contaminants deposit in the turbine and may be troublesome for runs over ten hours duration. When continued running is required, the turbine can be cleaned by injecting a mild abrasive (spent refinery catalyst or ground walnut shells) into the combustion system under load (Ref. 3-74). However, this will raise the maintenance cost of the plant, by 50 to 150 percent in comparison with GT No. 2 fuel operation. The pretreatment and contaminant removal add to the effective cost of the fuel. According to a study conducted in 1971 (Ref. 3-123), the cost of residual oil pretreatment would exceed the cost differential between the No. 3 GT and No. 4 GT fuels. However, this conclusion may require reappraisal in today's high priced fuel market.

It is clear, however, that treatment of residual oils may be another factor in limiting their use in heavy-duty gas turbines. One of the possibilities of making the use of residual oils more attractive is to remove the impurities in the refining process. Fuel desulfurization, which at present could be done most effectively at the refinery, will also remove most of the trace metals, hence eliminating or reducing the pretreatment requirements.

The effect of fuel bound nitrogen on NO_x was discussed in Section 3.3.3.2.2.1 and test data in Figure 3-47 show the exhaust NO_x concentration as a function of fuel-bound nitrogen. The data emphasize the NO_x problem with heavy and residual fuels.

The emission of SO_2 is directly related to the sulfur content in the fuel. Some means of SO_2 removal are discussed in Section 4.3.5.

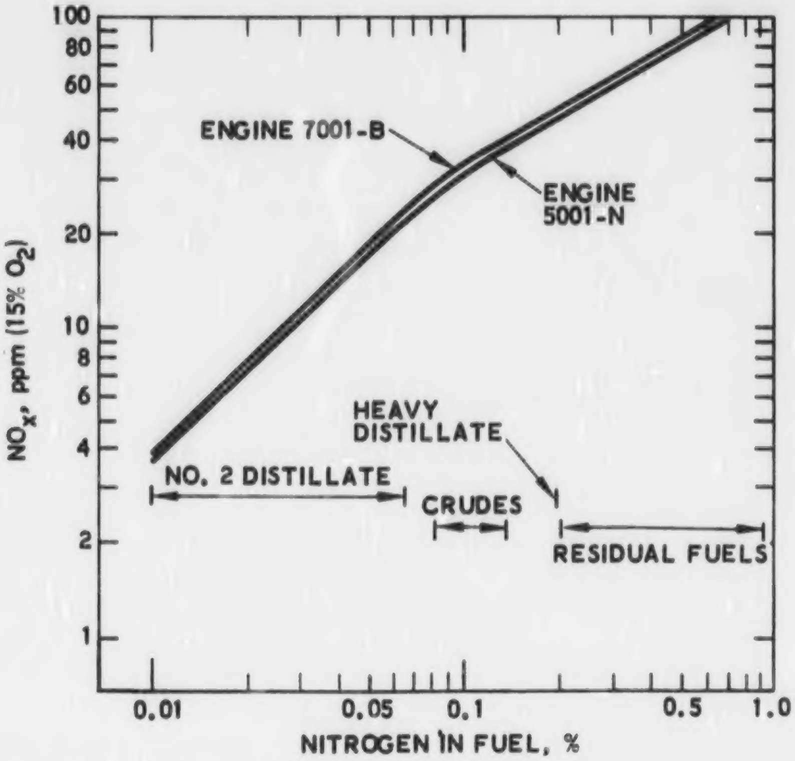


Figure 3-47. Effect of fuel bound nitrogen on NO_x formation at base load (Ref. 3-126)

The smoke formation with heavy fuels and crudes does not appear to be a problem and Refs. 3-123 and 3-124 report satisfactory operation with these fuels. The smoke measurements show no visible difference in comparison with No. 2 GT fuel due, probably, to preheating of the heavy fuels for good atomization.

Other means of smoke reduction were investigated in the past in the form of fuel additives and, for instance, Ref. 3-111 reported "drastic smoke reductions" of No. 2 GT fuel with 25 to 50 ppm manganese addition. However, the effect of this additive on turbine life was not investigated and better techniques such as improved combustor air flow distribution and atomization have since been adopted for smoke control.

Extended gas turbine operation on crude oils (1.3-million hours) and on residual oils (3.5-million hours) performed by one gas turbine manufacturer (Ref. 3-125) indicates no major technical problems. The control of emissions will be more critical with these fuels requiring careful attention to the combustor design and fuel atomization and, in the case of residual oils, more economical means of fuel pretreatment. Operation with low Btu coal gas should not present any problem since gas turbines have been operating successfully on blast furnace gas of 70 Btu/scf (Ref. 3-122).

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SECTION 4

AUTOMOTIVE EMISSION CONTROL TECHNOLOGY

This section reviews the various emission control techniques, devices, and systems which have been or still are under investigation by the automotive industry.

The presently known emission control approaches applicable to reciprocating and gas turbine engines can be divided into two categories: preventive and corrective. The preventive control methods, which include modifications of the operating conditions of the engine as well as certain engine design changes, are aimed at minimizing the formation of pollutants in the combustion chamber. Conversely, the corrective control methods include certain add-on devices, which are designed to convert the pollutants emitted from the engine to harmless compounds before admission into the atmosphere.

Potential emission control techniques for diesel engines are discussed in Subsection 4.1. Gasoline engines are treated in Subsection 4.2 and gas turbines are treated in Subsection 4.3.

4.1 DIESEL ENGINES

This section of the report deals with the work performed by many investigators to reduce the exhaust emissions from diesel engines and presents the most significant results achieved to date. Although a limited amount of pertinent test data have been reported for large stationary diesel engines, most of the research and development work concerned with emission control of diesels was performed on engines designed for use primarily in on-highway applications. Emission standards for this engine category have been in existence for some time, and more stringent standards might be promulgated by the EPA for future model year heavy-duty atuomotive engines.

The emission control techniques considered to date for use in diesel engines fall into one of the following four categories:

1. Variation in engine operating conditions
2. Engine component modifications
3. Incorporation of emission control devices
4. Combined systems.

The variations in engine operating conditions include engine load and speed; intake air temperature and pressure; fuel injection timing; and fuel characteristics. The effect of these parameters on emissions and fuel consumption is discussed in Section 4.1.1. The engine component modifications are described in Section 4.1.2, including changes in the combustion chamber geometry, compression ratio, valve timing, air swirl, and fuel injection system. The emission control devices evaluated to date include exhaust gas recirculation; water induction or injection; catalytic converters; thermal reactors; exhaust gas scrubbers; as well as incorporation of a turbocharger. These particular devices are examined in Section 4.1.3. Finally, Section 4.1.4 presents data from a number of emission control systems, consisting of combinations of various control devices/techniques.

4.1.1 Variations in Operating Conditions

4.1.1.1 Engine Load and Speed

The effects of varying engine load and speed on the HC, CO, and NO_x mass emissions of a four-stroke, turbocharged, divided-chamber diesel engine are illustrated in Figure 4-1 (Ref. 4-1). As indicated in the figure, a reduction in engine load at a given speed tends to increase the specific mass emissions of HC and NO_x, while reducing CO. Conversely, at constant load, a reduction in engine speed is accompanied by a marked improvement in the specific mass emissions of HC and NO_x, and a substantial rise in CO, except for the very low-load regime where the trends are reversed. Derating the engine from its rated operating point (285 bhp at 2200 rpm) down to 200 bhp and 1500 rpm results in a reduction in NO_x from about 4.3 g/bhp-hr to about 2.5 g/bhp-hr. This is accompanied by a 60-percent reduction in HC and a 150 percent increase in CO. As discussed in Section 3.1.4, similar trends have been observed on other diesels, while some engines indicate inverse relationships. These differences in trends are attributed to variations in the component design details and fuel injection versus engine load/speed schedules.

Although engine derating appears to be beneficial from a NO_x reduction point of view, at least for some engines, the related investment cost increase in terms of dollars per maximum brake horsepower and the associated changes in specific fuel consumption would have to be taken into account when considering this approach.

4.1.1.2 Intake Air Temperature and Pressure

Variations in the intake air temperature and pressure have been shown to have some effect on the emissions and specific fuel consumption of diesel engines. In general, NO_x is expected to decrease with decreasing charge temperature. This trend is attributed to the associated reduction in the compression and local reaction temperatures

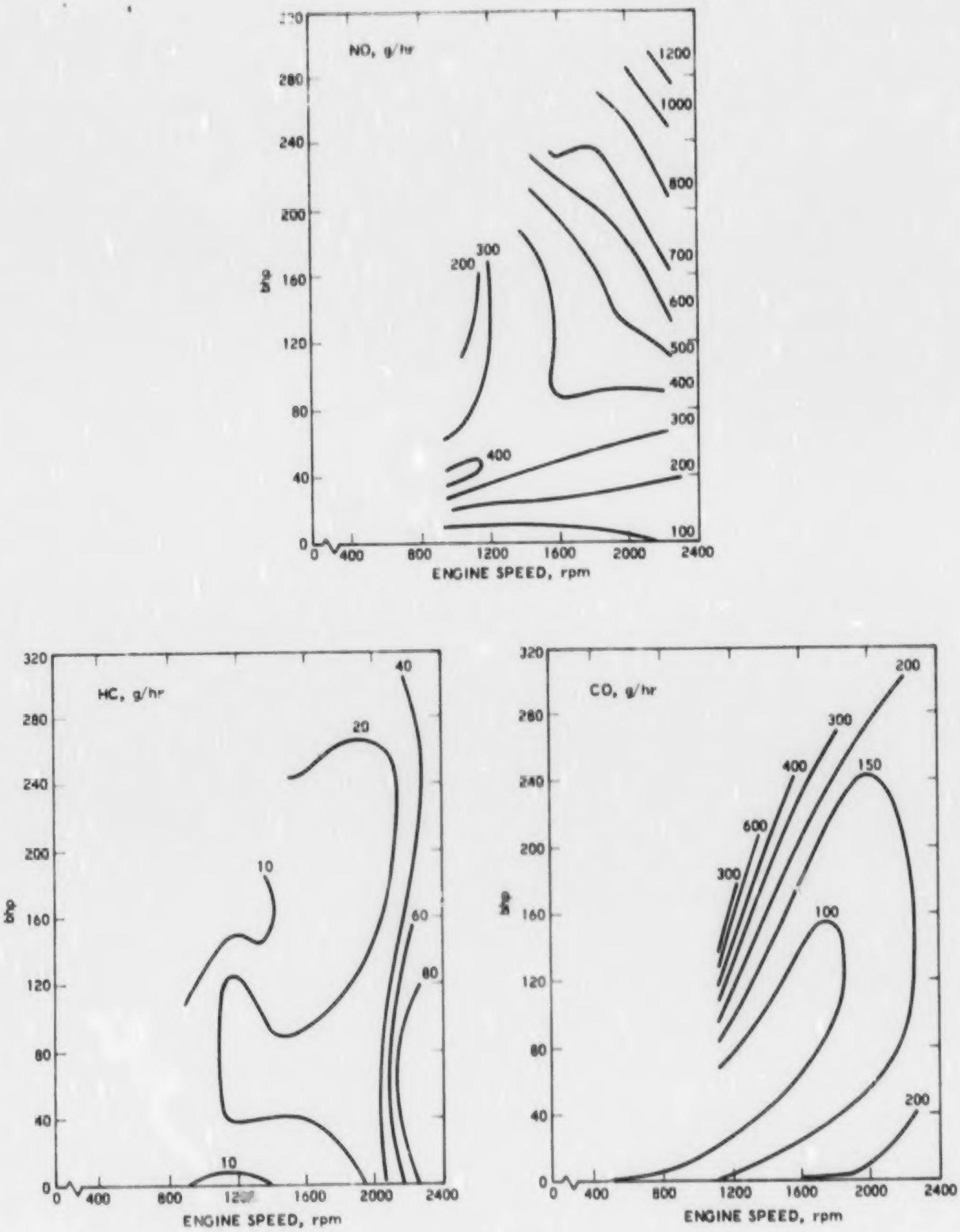


Figure 4-1. Effect of speed and power output on emissions - Caterpillar four-stroke precombustion chamber diesel (Ref. 4-1)

and NO_x formation rates. Moreover, a reduction in charge temperature tends to increase the ignition delay in the combustion chamber, which can further reduce NO_x.

The primary effects related to intake air throttling are mixture enrichment and reduction in the air mass flow rate. Based on theoretical considerations and the available test data (Ref. 4-2), it appears that little benefit could be derived from this particular technique.

4.1.1.2.1 Intake Air Temperature

The effect of intake air temperature on the NO_x emissions of a turbocharged, open-chamber diesel engine, rated at 230 bhp, is shown in Figure 4-2 (Ref. 4-2). In these tests, the engine was operated at its rated speed of 2100 rpm and the intake air temperature was controlled by means of a water-cooled heat exchanger installed between the turbocharger and the engine intake manifold. As expected, NO_x decreased substantially with decreasing manifold temperature, and the rate of reduction was essentially independent of engine load. Apparently,

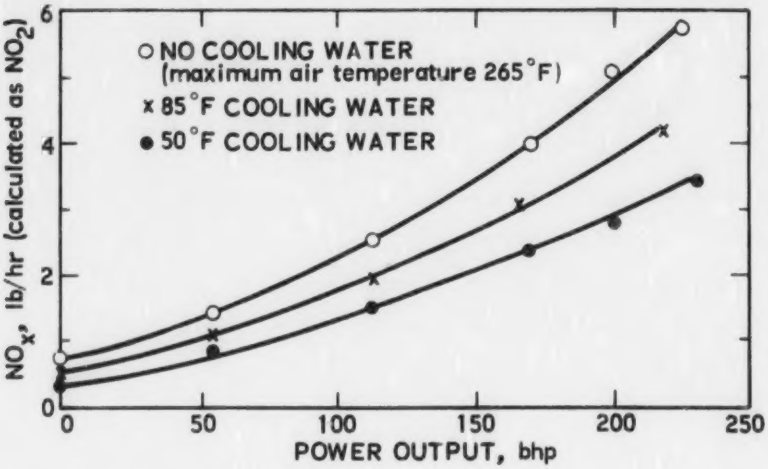


Figure 4-2. Effect of intake-air temperature on NO_x emission of a turbocharged, open-chamber diesel engine at rated speed (Ref. 4-2)

cooling of the intake air had no detrimental effect on the CO, HC, and smoke emissions of this engine. Also, fuel economy and odor remained essentially unchanged. With 50°F cooling water, the NO_x emissions of the engine were reduced by about 40 percent. Although cooling to 50°F might not be practical for most applications, it is evident that this approach is quite effective in reducing NO_x. Particularly in stationary engines, where the volume constraints are less severe, incorporation of a high effectiveness aftercooler might prove to be a cost effective NO_x abatement technique.

Test data from a turbocharged, divided-chamber diesel engine (No. 24 in Table 3-5) indicate that changing the air temperature from 250 to 150°F resulted in a 20 percent reduction in NO_x, accompanied by a four-percent improvement in specific fuel consumption. The HC, CO, and smoke emissions had a tendency to increase slightly with decreasing intake air temperature. At half load, the data showed considerable scatter, but there is some indication that the effectiveness of charge cooling with respect to NO_x reduction diminishes in the part-load regime (Ref. 4-3).

Similar NO_x reduction rates were obtained by the Bureau of Mines on a turbocharged, open-chamber diesel engine (No. 16 in Table 3-3) which was tested with air intake temperatures of 250 and 150°F. However, in this case the indicated improvement in specific fuel consumption was only about one percent. Data published by Bascom, et al. (Ref. 4-4) indicate a reduction in NO_x of about 30 percent by lowering the intake air temperature from 250 to 150°F. Somewhat smaller improvements were reported by Wilson, et al. (Ref. 4-5), for single-cylinder, open-chamber, and prechamber diesel engines. Reduction in the intake air temperature from 200°F to 100°F resulted in a 10 to 15 percent improvement in NO_x, accompanied by a slight gain in specific fuel consumption. At the same time, the smoke increased somewhat in the prechamber engine, but remained unchanged in the open-chamber engine.

Data from a large turbocharged diesel engine, rated at 4300 bhp, are presented in Table 4-1 (Ref. 4-6). This particular engine was operated at its design conditions (600 rpm and 200 psi BMEP) using both No. 2 diesel fuel and natural gas. With diesel fuel, reduction in the manifold temperature from 130°F (standard setting) to 100°F resulted in a five-percent reduction in NO_x and a concomitant 1.5-percent improvement in specific fuel consumption. In the case of natural gas, NO_x was reduced by 20 percent, whereas specific fuel consumption showed a 0.5-percent increase. Only small variations were observed in the emissions of CO and HC.

4.1.1.2.2 Projected Benefits

Although intake air cooling is not a very powerful NO_x reduction technique, it can be implemented rather easily, particularly in the case of stationary engines where size considerations are frequently less significant. Since specific fuel consumption tends to decrease with intake cooling, this technique might prove to be particularly attractive in conjunction with other control methods, such as fuel injection timing retard and EGR, to compensate for the loss in engine efficiency generally encountered with these approaches. The performance characteristics of combined techniques are futher discussed in Section 4.1.4.

It is concluded that a 0.15 to 0.3 percent reduction in NO_x can be achieved per degree reduction in intake air temperature. This is accompanied by a 0 to 0.04 percent improvement in fuel economy. The effect of manifold temperature on HC, CO, smoke, and odor appears to be rather insignificant.

TABLE 4-1. EFFECT OF INTAKE AIR COOLING ON THE EMISSIONS AND SPECIFIC FUEL CONSUMPTION OF A COOPER BESSEMER KSV-12 DIESEL ENGINE (Ref. 4-6)

Fuel	rpm	Load	Manifold Temp, °F	Injection Timing	Emissions, g/bhp-hr			BSFC,[a] $\frac{\text{Btu}}{\text{bhp-hr}}$	Emission Ratios			$\frac{BSFC^a}{BSFC_0}$
					HC	CO	NO_x		$\frac{HC}{HC_0}$	$\frac{CO}{CO_0}$	$\frac{NO_x}{NO_{x_0}}$	
No. 2 Diesel	600	Full	130	Standard	0.13	3.85	10.99	6677	-	-	-	-
No. 2 Diesel	600	Full	100	Standard	0.17	3.82	10.54	6583	1.3	0.99	0.96	0.986
No. 2 Diesel	600	Full	130	-4°	0.21	4.45	9.31	6732	-	-	-	-
No. 2 Diesel	600	Full	100	-4°	0.20	4.09	8.71	6636	0.95	0.92	0.94	0.986
Natural Gas	600	Full	130	Standard	5.16	4.50	8.96	6340	-	-	-	-
Natural Gas	600	Full	100	Standard	5.28	4.25	7.27	6377	1.02	0.94	0.81	1.006
Natural Gas	600	Full	130	-4°	3.22	7.21	8.41	6410	-	-	-	-
Natural Gas	600	Full	100	-4°	3.20	6.45	6.63	6444	0.99	0.89	0.79	1.005

[a] Brake Specific Fuel Consumption

[b] Subscript zero refers to baseline conditions

4.1.1.3 Fuel Injection Timing

Injection timing is probably the single most important parameter affecting the emissions from diesel engines. Early injection timing tends to produce high combustion pressure rise rates and peak temperatures, particularly when combined with high injection rates. As a result, the formation of NO_x in the combustion chamber increases. Conversely, retarded timing minimizes the formation of NO_x. Generally, a timing retard has a small impact on other pollutant species (except for very late timing) but degrades fuel economy (Refs. 4-1 and 4-4).

4.1.1.3.1 Single Cylinder Engine Data

The effect of fuel injection timing variations on diesel engine emissions and performance was evaluated by Pischinger and Cartellieri, using a single-cylinder, open-chamber research engine with a compression ratio of 16.2 and an inlet swirl ratio of 2.6 (Ref. 4-7). As indicated in Figure 4-3, timing retard resulted in decreasing NO_x emissions at the expense of some increase in CO, smoke, and specific fuel consumption. Conversely, timing advance caused sharply higher NO_x and CO. The smoke intensity decreased somewhat with advanced timing, while HC remained essentially constant. With respect to the optimum setting, five-degree timing retard resulted in a 44-percent reduction in NO_x, an 86-percent increase in CO, a 50-percent increase in smoke, and a three-percent increase in specific fuel consumption. For three degrees retard, NO_x decreased by 31 percent, while SFC increased by 1.1 percent, CO by 45 percent, and smoke by 33 percent. This indicates that the first few degrees of timing retard are particularly effective from an SFC versus NO_x tradeoff point of view. The magnitude of the effect is dependent upon the baseline timing.

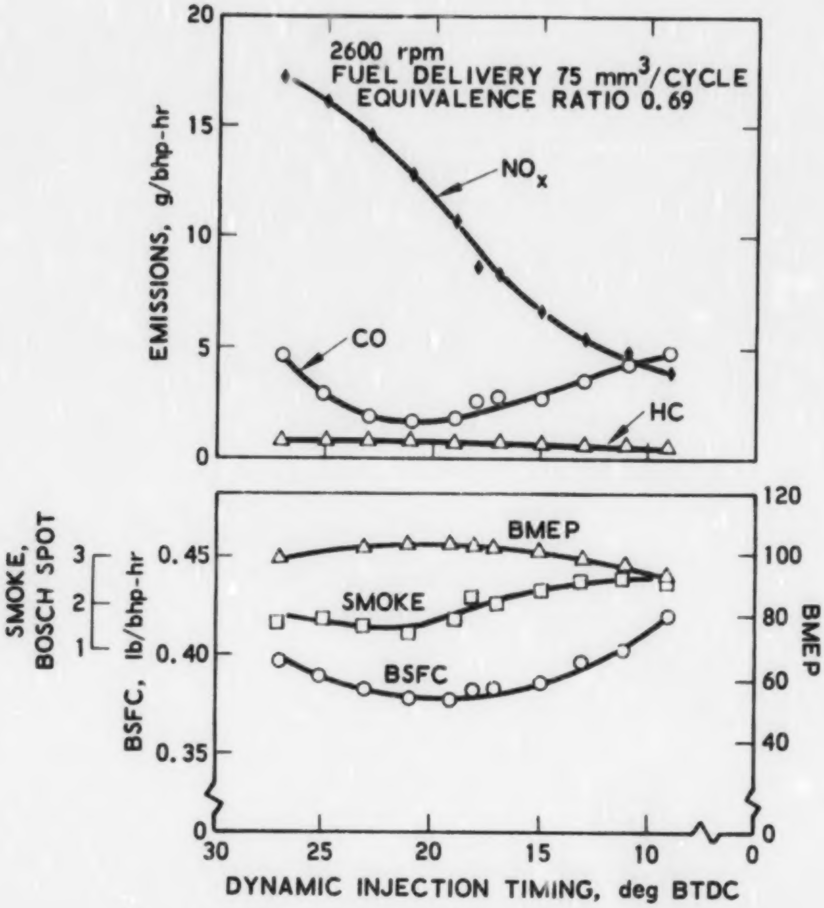


Figure 4-3. Effect of fuel injection on performance and emissions of a single cylinder research engine (Ref. 4-7)

Test data reported by Shahed et al. (Ref. 4-8), Wilson et al. (Ref. 4-5), and Valdmanis and Wulfhorst (Ref. 4-9) for single-cylinder, open-chamber diesel engines are summarized in Table 4-2, indicating somewhat lower NO_x reduction rates.

TABLE 4-2. SINGLE CYLINDER DIESEL ENGINE EMISSIONS AS A FUNCTION OF INJECTION TIMING RETARD

Timing, Degrees BTDC	Timing Retard, Degrees	$\frac{NO_x{}^*}{NO_{x_0}}$	$\frac{HC}{HC_0}$	$\frac{CO}{CO_0}$	$\frac{Smoke}{Smoke_0}$	$\frac{SFC}{SFC_0}$	Reference
20	0	1.0	-	-	-	0	4-8
17	3	0.76	-	-	-	1.007	
15	5	0.67	-	-	-	1.012	
12	8	0.57	-	-	-	1.030	
20	0	1.0	-	-	-	-	4-5
17	3	0.82	-	-	-	-	
15	5	0.69	-	-	-	-	
25	0	1.0	1.0	1.0	1.0	-	4-9
20	5	0.81	0.60	0.85	1.27	-	
15	10	0.61	0.23	0.71	1.33	-	

*Subscript zero refers to standard timing.

4.1.1.3.2 Multicylinder Four Stroke Naturally Aspirated Diesels

The effect of injection timing on the emissions of medium-speed, four-stroke naturally-aspirated, open-chamber diesel engines was investigated by Kahn et al. (Ref. 4-10) and by the Bureau of Mines (Ref. 4-3). Figure 4-4 reproduces the data published by Kahn for a 245 CID 4-cylinder engine. In these tests, the engine was operated at 2000 rpm (60 percent of rated speed) and full load, and the injection timing was varied from the standard setting of 20 degrees BTDC down to 0 degrees, using four discrete injection period settings (Systems 1 through 4). As indicated in the figure, the NO_x emissions and

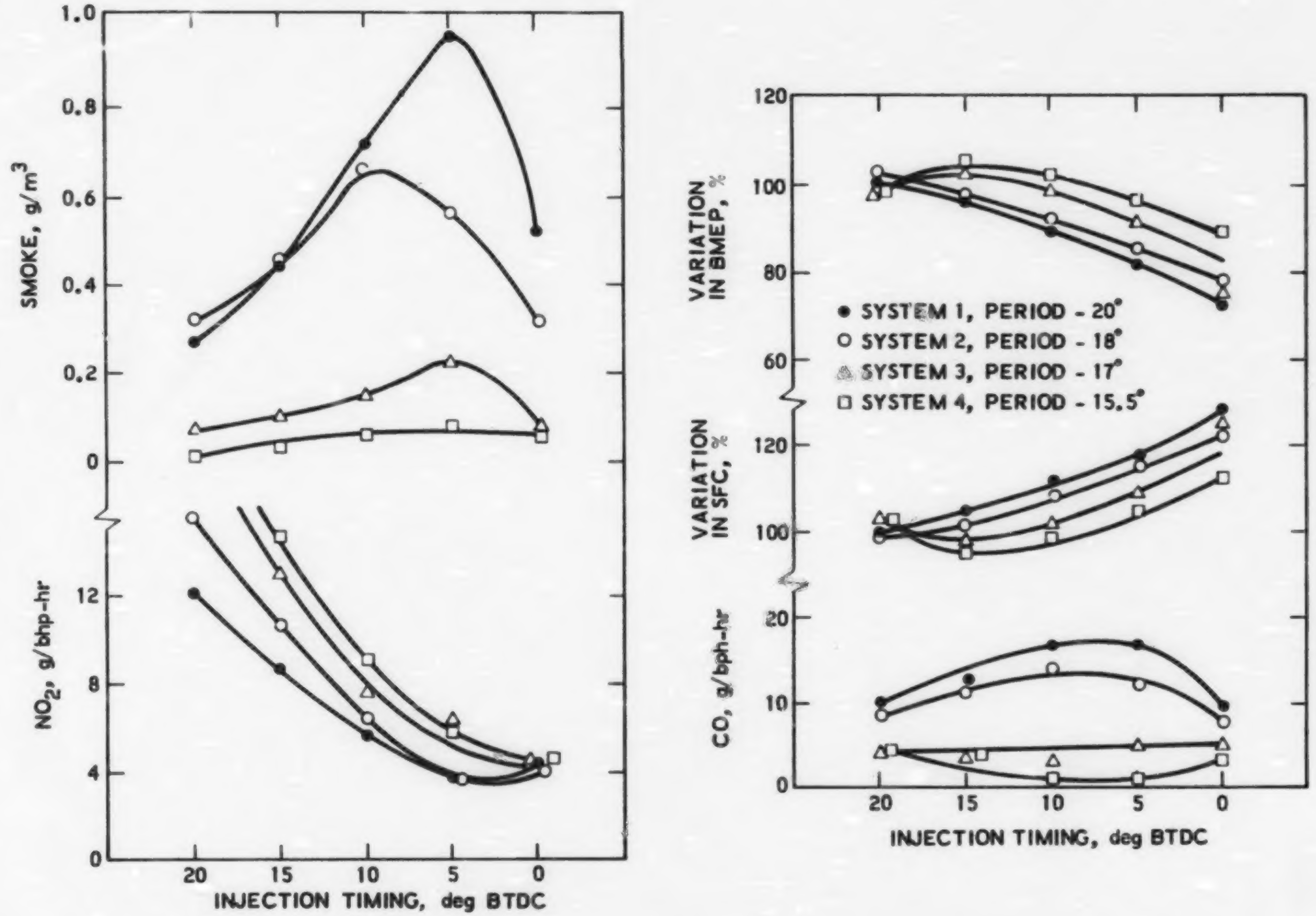


Figure 4-4. Effect of injection rate and timing on gaseous mass emissions, smoke, and performance of a naturally aspirated diesel engine (Ref. 4-10)

the power output capability of the engine decreased markedly with increasing timing retard, while smoke and specific fuel consumption increased. Not shown in the figure is HC, which remained essentially constant over the range of timing and injection period settings. Reducing the injection period resulted in higher NO_x emissions and lower SFC, CO, and smoke levels. At the standard injection period of 20 degrees, retarding the injection timing from 20 degrees BTDC to five degrees caused a change in NO_x from 12 g/bhp-hr to 4 g/bhp-hr, a reduction of 66 percent. However, this improvement was accompanied by an 18-percent loss in engine power (BMEP) and SFC and a substantial increase in CO and smoke. Conversely, with the shortest injection period of 15.5 degree, retarding the timing from 15 degree (which is optimum for this setting as far as fuel economy is concerned) to zero degree resulted in a reduction in NO_x from 14.5 g/bhp-hr to about 4.4 g/bhp-hr. In this case, the loss in SFC and power output is only about 12 percent relative to the standard setting (System 1), and the smoke and CO emissions are reduced also.

Curves computed from test data published by the Bureau of Mines (Ref. 4-3) are presented in Figures 4-5 and 4-6, showing engine performance and mass emission parameters of a four-stroke, naturally aspirated, open-chamber diesel as a function of injection timing and load/speed settings. The standard timing of this engine is 34 degree BTDC and the data obtained at this point were utilized to normalize the parameters plotted in the graphs.

Except for the intermediate speed/full-load condition, the specific fuel consumption is a minimum at the standard injection timing. For a given operating condition, the exhaust smoke intensity varied very little in the retarded regime, but increased rapidly as timing was advanced from the standard setting. As expected, smoke intensity increased with increasing load. Also shown in Figure 4-5 is the maximum power output capability of the engine as affected by timing.

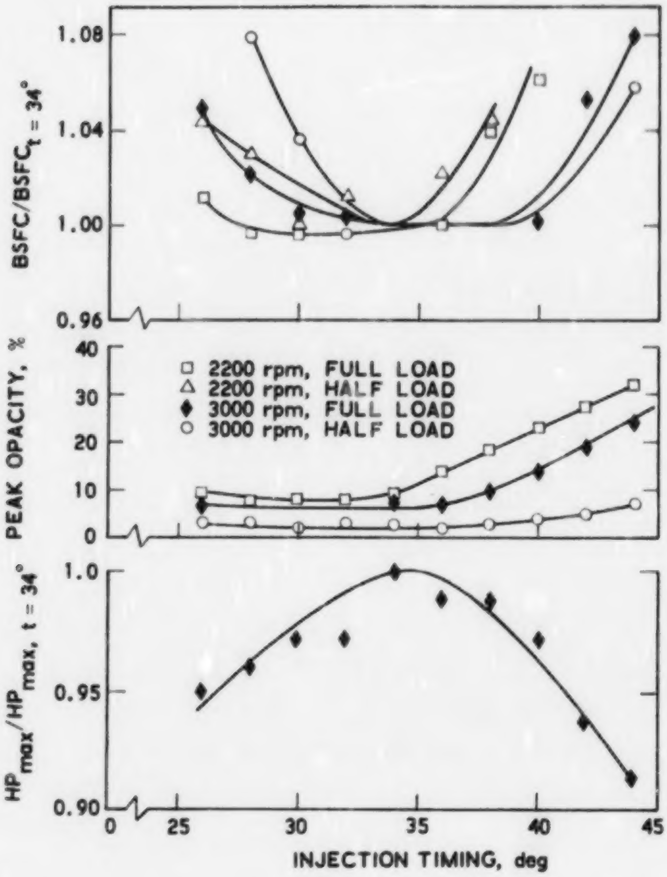


Figure 4-5. Effect of injection timing on specific fuel consumption, peak opacity, and maximum power output of a naturally aspirated, open-chamber diesel engine — Engine No. 7 (Ref. 4-3)

Retarding or advancing the timing degraded engine power somewhat. For instance, five degrees retard caused a three-percent loss in power. At full load, the HC emissions tend to decrease with increasing retard and show a substantial rise for advanced timing. At half load, these trends are reversed. CO varied moderately in the retarded regime,

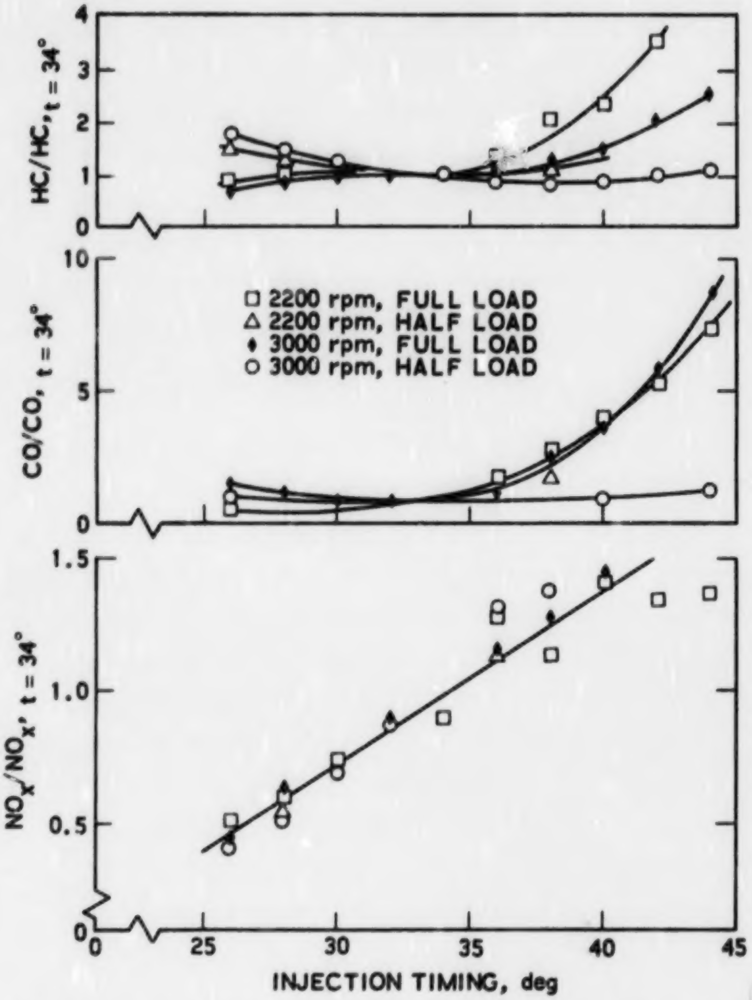


Figure 4-6. Effect of injection timing on the emissions of a naturally aspirated, open-chamber diesel engine - Engine No. 7 (Ref. 4-3)

but increased rapidly for advanced timing. NO_x shows a linear relationship with timing. Five degrees retard resulted in a 35-percent reduction in NO_x.

NO_x emission and performance data from another naturally aspirated engine tested by the Bureau of Mines (Engine No. 7 in Table 3-2) show a more rapid reduction in NO_x with increasing timing retard. For example, 5 degree retard caused a 50-percent reduction in NO_x, accompanied by a 7.6-percent increase in SFC and a 7-percent loss in engine power output capability. The HC and CO emissions of this engine showed considerable data scatter, but HC decreased with increasing timing retard, whereas CO showed a tendency to increase. These trends are in general agreement with the curves in Figure 4-6.

4.1.1.3.3 Multicylinder, Four Stroke, Turbocharged Diesels

A medium-speed, turbocharged, open-chamber diesel engine, rated at 325 bhp-hr (Engine No. 16 in Table 3-3) was tested by the Bureau of Mines with standard injection timing and with 3 degree retard (Ref. 4-3). At the retarded setting, the NO_x emissions at full load and at half load were 25 percent lower than for zero retard. This reduction was accompanied by a 2.7-percent increase in specific fuel consumption and a one-percent loss in maximum engine power. At the same time, the peak smoke opacity of the engine increased from 8.6 percent to 12.2 percent, whereas HC and CO showed only moderate variations.

One manufacturer of medium-speed diesel engines has provided test data showing a 30-percent reduction in NO_x and a four-percent loss in specific fuel consumption when operating with 5 degree retard.

Data published by Cooper-Bessemer for a large-bore diesel engine tested with both No. 2 diesel fuel and natural gas are listed in Table 4-3 (Ref. 4-6). As indicated, retarding the timing by 4 degrees resulted in a 15-percent reduction in NO_x for No. 2 diesel fuel and a six-percent reduction for natural gas. In both cases, the specific fuel consumption of the engine increased by about one percent, while the observed changes in HC and CO were relatively small.

TABLE 4-3. EFFECT OF 4° INJECTION TIMING RETARD ON EMISSIONS AND FUEL CONSUMPTION OF A LARGE STATIONARY DIESEL (Ref. 4-6)

Fuel	rpm	Load	Injection Timing	Mass Emissions, g/bhp-hr			BSFC,[a] $\frac{\text{Btu}}{\text{bhp-hr}}$	Emission Ratios			$\frac{BSFC^a}{BSFC_0}$
				HC	CO	NO_x		$\frac{HC}{HC_0}$[b]	$\frac{CO}{CO_0}$	$\frac{NO_x}{NO_{x0}}$	
No. 2 Diesel	600	Full	Standard	0.13	3.85	10.99	6677	-	-	-	-
No. 2 Diesel	600	Full	-4°	0.21	4.45	9.31	6732	1.61	1.16	0.85	1.01
Natural Gas	600	Full	Standard	5.16	4.50	8.96	6340	-	-	-	-
Natural Gas	600	Full	-4°	3.22	7.21	8.41	6410	0.62	0.94	0.94	1.01

[a] Brake Specific Fuel Consumption

[b] Subscript zero refers to standard timing

Test data from another large-bore, turbocharged, open-chamber diesel engine are plotted in Figure 4-7 as a function of injection timing retard. Comparison of these data with Table 4-3 indicates that varying injection timing on this engine had a larger effect on NO_x than in the case of the Cooper-Bessemer engine. For example, retarding the timing by four degrees from the standard setting caused a change in NO_x from 12.5 g/bhp-hr to 8.5 g/bhp-hr, a reduction of 32 percent. However, the NO_x improvement was associated with a 4.2-percent loss in specific fuel consumption.

4.1.1.3.4 Two Stroke Diesels

The effect of injection timing variations on NO_x, CO, and BSFC of a General Motors 6-71N diesel engine is illustrated in Figure 4-8 (Ref. 4-11). Since engine flow rate remained nearly constant at each speed, the mass emissions were proportional to the concentrations shown in the figure.

For both speeds, NO_x decreased with increasing timing retard, but increased with engine load due to the higher temperature levels in the combustion chamber. At peak torque speed, NO_x reached a maximum at about 90-percent load and then decreased again, indicating a deficiency in available oxygen. At 2100 rpm, NO_x decreased by about eight percent per degree injection retard, and at 1200 rpm, the rate of reduction was about seven percent per degree. Within the range of acceptable smoke levels, the effects of timing changes on HC and CO were not very significant. However, when timing was retarded beyond these levels, CO increased sharply at full loads, while HC increased at light loads, indicating incomplete combustion. As shown in Figure 4-7, timing retard has a detrimental effect on SFC. For example, five degrees retard resulted in a 3.5-percent loss in SFC. These rates of change are in reasonable agreement with the previously discussed four-stroke engines.

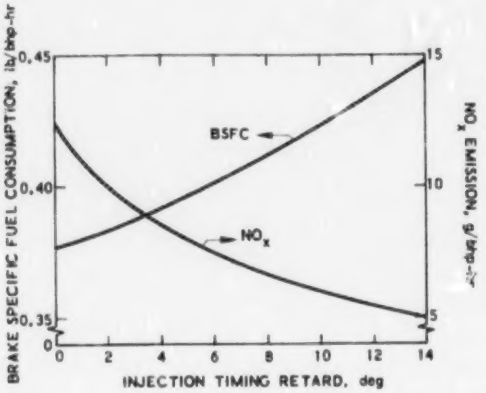


Figure 4-7. Effect of injection timing retard on the NO_x emissions and specific fuel consumption of a large diesel engine

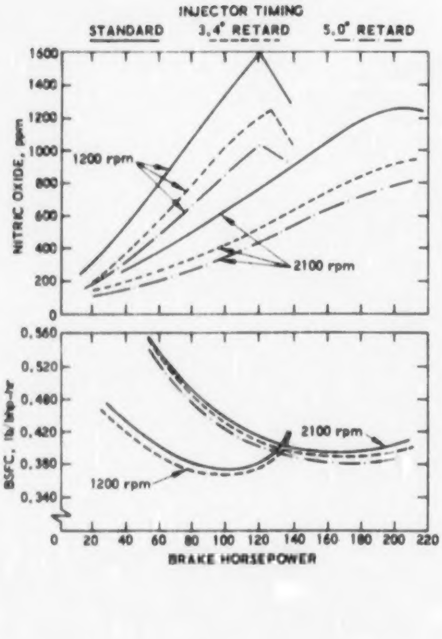


Figure 4-8. Effect of injection timing on NO_x emissions and specific fuel consumption of a two-stroke diesel engine (Ref. 4-11)

4.1.1.3.5 Proposed NO_x Reduction Correlations

The NO_x reductions determined from the engine tests discussed in the preceding sections are summarized in Figure 4-9 as a function of timing retard and specific fuel consumption degradation. The medium-speed, multicylinder engines are distributed almost uniformly over the NO_x versus SFC data band, whereas the single-cylinder engines are concentrated in the upper half of the band and the large, low-speed engines in the lower half. Conversely, in the NO_x versus timing chart, most of the single-cylinder engine data fall within the lower half of the band, while the majority of the medium-speed, multicylinder engines is included in the upper half.

The available test data relating the effect of injection timing retard and the variations in HC, CO, and smoke are shown in Figure 4-10. In some engines, HC and CO increase markedly with increasing timing retard, but remain constant or decrease in others.

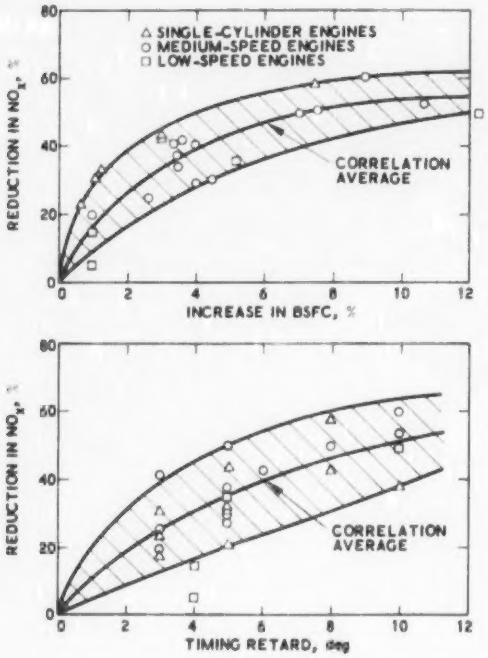


Figure 4-9. NO_x reduction in diesel engines vs specific fuel consumption and timing retard

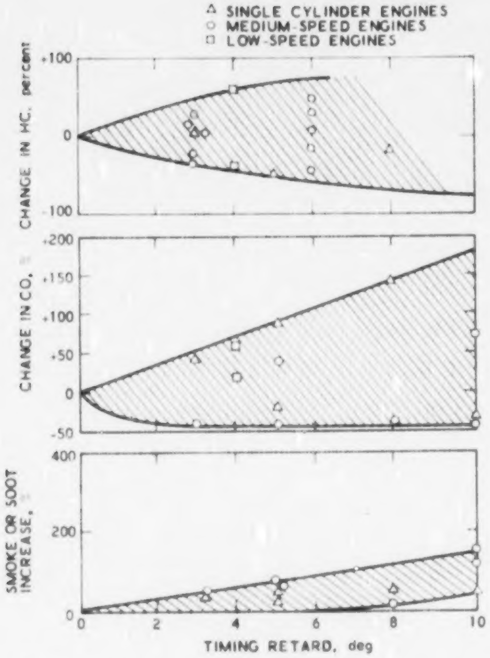


Figure 4-10. HC, CO and smoke variations vs timing retard

Smoke tends to increase as timing is retarded, but the magnitude of the increase seems to vary widely.

4.1.1.4 Fuel Effects

Diesel engine emission test work conducted to date, using different distillate type fuels, has indicated that variations in the fuel can produce small changes in the emissions. However, the potential improvements related to variations in fuel properties cannot always be predicted and are often affected by the design of the engine, as well as its operating conditions (Ref. 4-12). In some cases, changes in certain fuel properties affect the exhaust emissions by altering the engine operating characteristics rather than by changing the combustion process directly. Thus, a change in engine design or its mode of operation might achieve the desired result more easily than would a revision of fuel quality.

On the other hand, utilization of heavy fuels might cause higher NO_x emissions as a result of the conversion of the fuel-bound nitrogen to NO. Test data indicate that up to 70 percent of the fuel-bound nitrogen is converted in steam boilers (Ref. 4-13), but information is lacking regarding the conversion in diesel engines.

4.1.1.4.1 Cetane Number and Composition

The effect of fuel cetane number on the HC, CO, and NO_x emissions of a number of open-chamber, naturally aspirated and turbocharged diesel engines is shown in Figure 4-11 (Ref. 4-14). As indicated, the HC, CO, and NO_x emissions decrease moderately with increasing fuel cetane number, particularly in the case of the two naturally aspirated engines.

Similar results were reported by Cummins for a single-cylinder experimental diesel engine which was operated with three different fuels having cetane ratings of 40, 52, and 56, respectively (Ref. 4-8). As shown in Figure 4-12, the NO_x specific mass emissions decrease with increasing cetane number over the whole range of fuel injection timing settings investigated. Increasing the cetane number of the fuel resulted in a more favorable tradeoff between fuel consumption and NO_x emissions.

The influence of fuel composition on the emissions of several four-stroke, naturally aspirated and turbocharged open-chamber, and two-stroke, open-chamber, air-scavenged diesel engines has been studied by the Bureau of Mines (Ref. 4-2). The properties of the seven test fuels used in the program covered wide ranges: specific gravity varied between 0.81 and 0.87, the aromatic content varied between 14 and 44 percent, the cetane number varied between 50 and 42, and the sulfur content varied between 0.04 and 0.4 percent. The emission variations obtained with these different fuels were quite small, except for CO which, at full-load conditions, increased markedly with

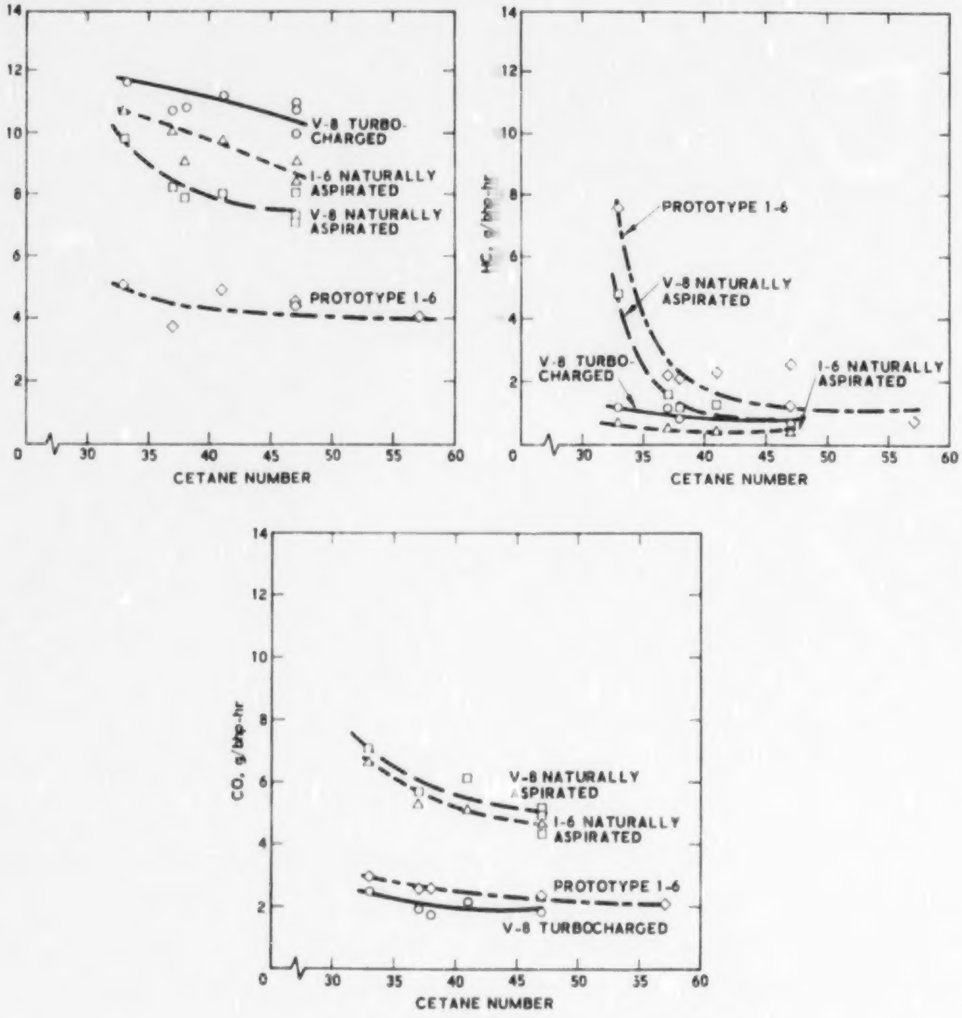


Figure 4-11. Cetane number effects on the emissions of naturally aspirated and turbocharged diesels (Ref. 4-14)

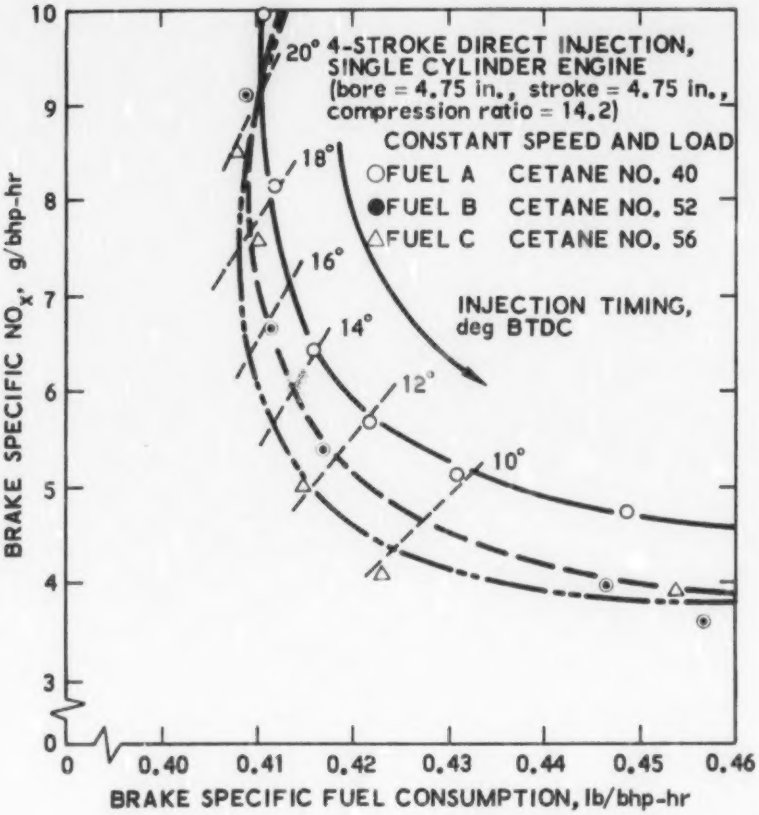


Figure 4-12. Measured effect of fuel cetane number on NO_2 vs BSFC tradeoff (Ref. 4-8)

increasing fuel gravity, owing to the associated reduction in air-fuel ratio. Examination of the test data shows that HC and NO_x increased slightly with decreasing cetane number. Odor intensity was affected primarily by the sulfur content in the fuel and increased by about 20 percent as the sulfur content was raised from 0.04 percent to 0.4 percent.

There is some indication that exhaust smoke opacity increases with increasing cetane number (Refs. 4-4 and 4-15). This trend is attributed to the lower stability of the higher cetane fuels.

Conversely, an increase in fuel volatility has resulted in reduced smoke intensity (Ref. 4-16).

4.1.1.4.2 Smoke Suppressant Additives

A number of smoke suppressant fuel additives have been evaluated by several investigators. Basically, these additives can be divided into two groups. One group is of the detergent type and is designed to maintain a clean fuel injection system. The second group consists of metal-based materials which reduce the ignition temperature of the carbon and promote its oxidation.

Barium-based fuel additives have been used successfully by a number of investigators to reduce diesel smoke intensity (Refs. 4-4 and 4-17 through 4-19). Test data reported by Bascom et al. (Ref. 4-4) for a four-stroke, naturally aspirated engine operated with a commercially available barium-based fuel additive indicate a reduction in smoke intensity from about 9 mg/ft^3 to less than 1 mg/ft^3 when 0.5 percent (by volume) of the additive was used. However, after 30 hours of testing with the treated fuel, the HC emissions had increased considerably, whereas CO and NO_x remained unchanged. The increase in HC was not due to an increase in deposit buildup in the engine, converse to what was observed by General Motors (Ref. 4-20).

An extensive study of the effects of fuel additives on diesel smoke was performed by Saito and Nabetani (Ref. 4-18), using two- and four-stroke open-chamber and prechamber engines. The The barium-containing additives proved to be most effective, resulting in a reduction in smoke opacity between 35 and 60 percent. In general, the effectiveness of the additive increased with increasing barium content. In these tests, the additives had essentially no effect on HC and showed a tendency to increase CO and to decrease NO_x.

Tests conducted earlier by the Bureau of Mines support these findings. However, after 100 hours of testing, the smoke level on one of the engines used in this program started to increase and the level obtained with the untreated fuel was approached after 100 additional hours of testing. This increase was attributed to deposit buildup in the combustion chamber and the injection nozzles (Ref. 4-19).

Approximately 70 percent of the barium added to the fuel is exhausted from the engine in the form of harmless barium sulfate, whereas part of the remainder consists of soluble compounds of which some are toxic. Although the barium-containing additives are quite effective in reducing diesel smoke levels, they are not recommended by most engine manufacturers because of adverse effects on engine durability.

4.1.1.4.3 Odor Effects

Although fuel composition often is suggested to have a prominent influence on diesel odor, the supporting experimental evidence is not strong. This conclusion is supported by tests conducted by the Bureau of Mines (Ref. 4-19). In these tests, a two-stroke diesel engine was operated with seven different diesel fuels which had different sulfur and aromatic contents. Although some differences in exhaust odor level were noted, the variations could not be reliably related to any one fuel property.

4.1.2 Component Modifications

4.1.2.1 Combustion Chamber

Modifications in the design of the combustion chamber are known to influence the fuel-air mixing and combustion processes occurring in diesel engines, and, as a result, the emission and specific fuel consumption characteristics of these engines are altered. The design parameters evaluated by a number of engine manufacturers include the geometry of the combustion chamber, compression ratio, valve timing, and air swirl. These factors are briefly discussed in the following paragraphs.

4.1.2.1.1 Chamber Geometry

The effect of variations in the ratio of combustion chamber bowl diameter to cylinder bore on the emissions of a single-cylinder, open-chamber research engine was evaluated by Pischinger and Cartellieri (Ref. 4-7). Increasing the bowl-diameter-to-bore ratio from 0.51 to 0.62 resulted in moderate reductions in NO_x and HC without affecting specific fuel consumption. However, the exhaust smoke level of the engine was substantially higher for the larger bowl diameter. The observed changes are attributed to better fuel and temperature distributions in the combustion chamber achieved with the larger bowl design.

Results from a combustion chamber shape study provided by one engine manufacturer are presented in Figure 4-13, showing specific fuel consumption and NO_x and HC emissions over the 13-mode test cycle. Again, NO_x decreases with increasing bowl size. Also, NO_x and HC emissions and fuel consumption vary slightly with piston head clearance (Ref. 4-14).

Based on the available data, it is concluded that variations in the bowl-diameter-to-bore ratio can have a small but distinct effect on NO_x, HC, and engine efficiency. Although the magnitude of the effect is expected to vary from engine to engine, optimization of this parameter can be beneficial as far as the tradeoff between emissions and fuel consumption is concerned.

In prechamber engines, the ratio of prechamber volume to total cylinder volume, the size of the orifice or orifices connecting the two chambers, and the internal shape of the prechamber have been found to have some effect on the NO_x and HC emissions (Ref. 4-1). Test data indicate that the NO_x and smoke emissions from these engines

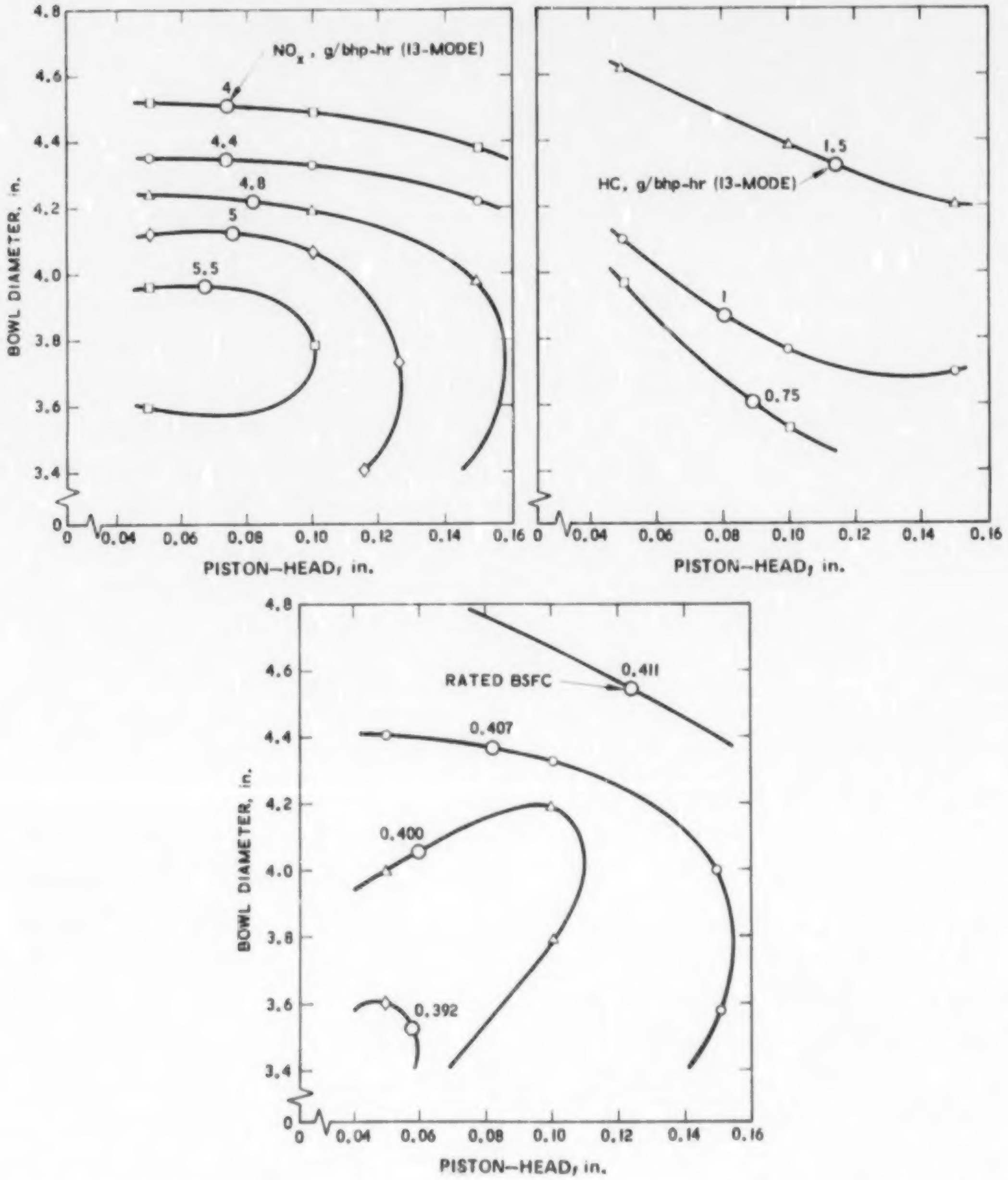


Figure 4-13. Effect of bowl diameter and piston head clearance on diesel engine emissions and specific fuel consumption (Ref. 4-14)

increase with increasing prechamber size, whereas specific fuel consumption shows some improvement (Ref. 4-5). These trends are plausible, because more air is burned in the prechamber as its size increases, thus reaching higher flame temperatures and NO_x levels.

4.1.2.1.2 Compression Ratio

The effects of engine compression ratio on the NO_x and HC emissions of an open-chamber diesel engine are shown in Figure 4-14. The HC emissions declined significantly with increasing compression ratio, while NO_x decreased slightly up to a compression ratio of about 16:1 and remained almost constant beyond that point. Since high compression ratios result in higher mechanical engine loads, the practical limit in compression ratio is dependent upon the BMEP and the engine structure but is about 18:1 for large diesel engines. Also, the small clearance volumes associated with high compression ratios inhibit the fuel-air mixing process, hence causing a loss in specific fuel consumption.

These trends are in agreement with the data reported by Wilson, et al., for a single-cylinder prechamber diesel engine (Ref. 4-5). However, Wilson's test data for an open-chamber engine indicate increasing NO_x emissions as the compression ratio of the engine was raised from 14 to 17.

4.1.2.1.3 Valve Timing

Valve timing, which affects cylinder scavenging and combustion efficiency, has been shown to have some influence on the emissions from diesel engines. In the past, the engines were designed for minimum fuel consumption and thermal loading of internal parts. However, in future engines, a lower scavenging efficiency might be selected to provide some internal exhaust gas recirculation (EGR) for the purpose of reducing NO_x (Ref. 4-1). The effectiveness of this

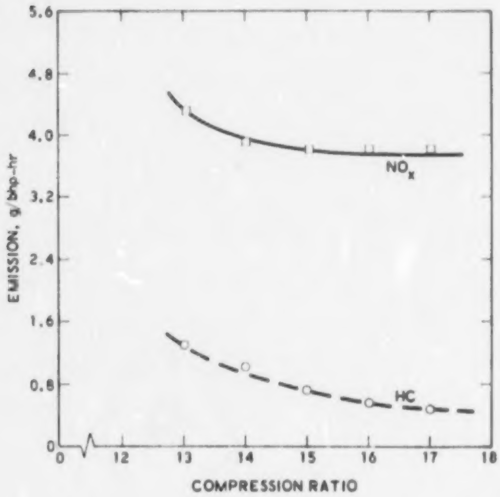


Figure 4-14. Effect of compression ratio on NO_x and HC emissions (Ref. 4-14)

approach would have to be compared with other potential emission control techniques, such as timing modifications, EGR, or water injection, before its application would be warranted.

4.1.2.1.4 Air Swirl

Air motion in the combustion chamber has important effects on the combustion process occurring in diesel engines and hence, emissions. In general, increasing the air swirl reduces the exhaust smoke intensity, but tends to increase NO_x. This trend is attributed to the associated improvement in fuel-air mixing which increases the reaction rates in the combustion chamber, causing a rise in the local temperatures and NO_x formation processes (Refs. 4-4 and 4-21).

Air swirl is particularly important in naturally aspirated engines, which often are operating at the smoke-limited air-fuel ratio. Conversely, turbocharged diesels appear to be less sensitive to air motion, probably because of the high excess air ratio used in these engines. This is illustrated in Figure 4-15, showing an inverse relationship between smoke and NO_x. The open-chamber engine used in these tests was modified to reduce the smoke level by (1) increasing the air swirl, (2) varying injection timing, and (3) turbocharging (Ref. 4-4). At a given smoke density, the lowest NO_x emissions were achieved with turbocharging. Conversely, increasing air swirl resulted in unfavorable NO_x versus smoke tradeoffs.

Similar results were obtained by Khan et al. (Refs. 4-10 and 4-21) and Wilson et al. (Ref. 4-5) on single-cylinder, open-chamber diesels. Increasing the swirl ratio resulted in lower specific fuel consumption, smoke, and NO_x levels and made the smoke vs. timing curve less sensitive to timing changes. Since higher swirl accelerates combustion, it might be desirable to operate the engine with retarded timing, which could result in a net reduction in NO_x without sacrificing fuel economy (Ref. 4-21).

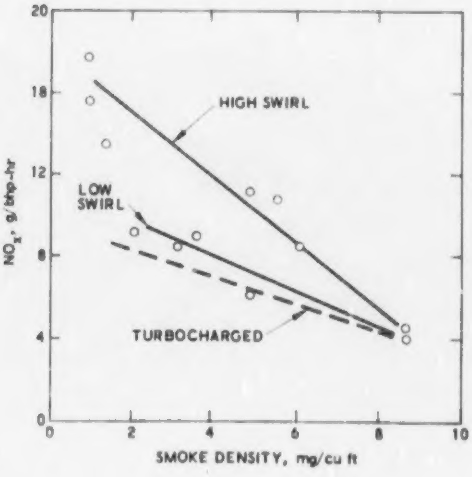


Figure 4-15. Effect of air swirl and turbocharging on smoke and NO_x emissions of open-chamber diesel engines (Ref. 4-4)

4.1.2.2 Injection System

4.1.2.2.1 Injection Characteristics

The design and operation of the fuel injection system has an important effect on the emissions from diesel engines — in particular, HC and smoke (Ref. 4-4). To minimize these emission species, the injection system must be capable of providing a uniform injection pulse and a properly distributed fuel spray without impingement on the piston or cylinder walls. Injector cup and spray geometry and injection duration are important design variables, and their effects on emissions are intimately related to the overall design of the combustion chamber (Ref. 4-4).

4.1.2.2.2 Sac Volume

Tests conducted by General Motors on a number of two-stroke diesel engines indicate that the sac volume at the tip of the needle valve was responsible for most of the relatively high HC emission obtained with one particular fuel injection system (Refs. 4-11 and 4-22). In this design, the fuel contained in the sac volume is boiled off after completion of the main injection event and is then discharged into the combustion chamber too late in the cycle to be fully oxidized. Reduction of the sac volume from the original 3.5 cubic millimeters to 0.5 cubic millimeters resulted in a reduction in HC from about 350 ppmC to about 90 ppmC. Further reduction to about 40 ppmC was achieved by means of an experimental needle valve design which covers the tip of the injector orifice in the closed-valve position. The aforementioned reduction in HC was obtained without sacrificing fuel economy, NO_x, and CO (Ref. 4-11). Similar improvements were realized by the Electromotive Division of General Motors on large locomotive diesels.

4.1.2.2.3 Injection Rate

The rate of injection has been shown to have a marked effect on the NO_x emissions of diesel engines. This is illustrated in Figure 4-16, showing the NO_x concentration in the exhaust of a two-stroke diesel, for three fuel injection rate settings (Ref. 4-11). In the high-load regime, a 20-percent reduction in NO_x was realized by increasing the rate of injection from 5.7 to 8.3 cubic millimeters per degree, and this effect was essentially independent of injection timing. At the same time, CO was reduced slightly, indicating improved combustion efficiency, while HC remained constant. Similar results were reported by Khan et al. (Ref. 4-10).

The principal drawback related to the use of shorter injection periods is the requirement of higher injection pressures, causing higher pushrod and cam loadings. As a result, engine durability might be adversely affected (Ref. 4-11).

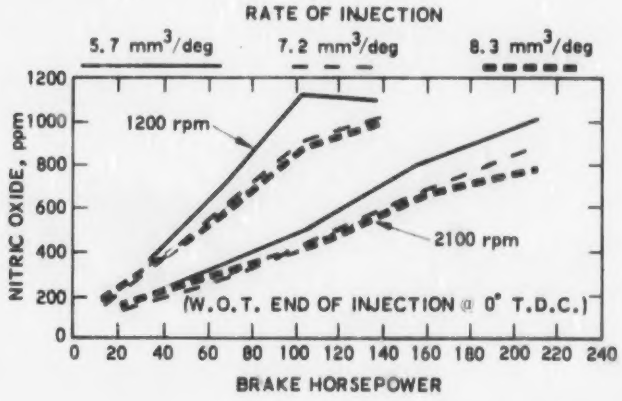


Figure 4-16. Effect of injection rate on NO_x emissions of a two-stroke diesel engine (Ref. 4-11)

4.1.2.2.4 Orifice Size and Spray Angle

Injection nozzle size affects the formation of the fuel spray and the fuel-air mixing process in the combustion chamber, and can have a significant impact on the emissions and performance of diesel engines. Test data from one engine indicate that a 20 to 40 percent reduction in NO_x was achieved by increasing the orifice diameter from 0.0055 inch to 0.0065 inch (without reducing the number of orifices). However, this improvement was accompanied by higher specific fuel consumption (three percent), CO emissions (70 to 100 percent), and smoke. These trends indicate that the observed reduction in NO_x was due primarily to lower combustion efficiency (Ref. 4-11). Reducing the number of injection orifices from eight to six, without changing the total orifice area, resulted in some reduction of NO_x at the expense of slightly higher fuel consumption, smoke, and CO emissions. Again, this is attributed to lower combustion efficiency caused by incomplete fuel-air mixing in the cylinder.

Similar effects were reported by Khan et al. (Ref. 4-10) and Wilson et al. (Ref. 4-5). The available data show that an optimum number and size of injection holes might exist for each particular engine design, hence permitting a tradeoff between the emissions and fuel economy.

Correlations showing the effects of injection hole number and spray angle on diesel engine emissions and fuel consumption were provided by one manufacturer (Ref. 4-14). Although the effects are not very large, optimization of these design parameters is desirable to minimize engine emissions. Apparently, the spray angle giving minimum fuel consumption and HC causes NO_x to reach a maximum.

4.1.2.2.5 Fumigation

Adding fuel to the intake air is known as fumigation. This technique has been used in the past to reduce the pressure rise rate in the combustion chamber and the noise of the engine. In the tests reported by Bascom et al. (Ref. 4-4), fumigation with kerosene increased the CO, HC, and smoke emissions of a naturally aspirated, open-chamber diesel engine while lowering NO_x slightly. With propane fumigation, the NO_x reduction effectiveness was improved, particularly at part loads.

4.1.2.2.6 Combined Effects

A number of the injector modifications discussed above were incorporated into a single injector which was then tested in a General Motors two-stroke diesel engine. The modifications included valve-covered orifices, higher injection rates, larger tip orifices plus retarded timing. Over the 13-mode test cycle, the following improvements in emissions were achieved relative to the standard engine settings. HC was reduced from 0.6 g/bhp-hr to 0.2 g/bhp-hr, NO_x was reduced from 9.0 g/bhp-hr to 5.0 g/bhp-hr, while CO increased from 3.6 g/bhp-hr to 4.8 g/bhp-hr. These changes were accompanied by a 2.5-percent loss in fuel economy and an increase in the exhaust smoke level by about one Bosch number.

The effect of injection system modifications on the emissions of a two-stroke, naturally aspirated, open-chamber diesel, rated at 280 bhp (engine No. 28 in Table 3-6), was investigated by the Bureau of Mines (Ref. 4-3). The engine was operated at its rated speed (2100 rpm), using the standard injectors and a modified injection system incorporating low sac volume, high injection rates, and a special timing schedule that provided for variable beginning and constant ending of injection. The test results indicated substantial reductions in HC and NO_x with the new injector, particularly in the low power regime.

These improvements were accomplished without compromising SFC, CO, and smoke. For example, near rated load, the specific mass emissions of NO_x and HC were reduced by about 25 and 60 percent, respectively. At half load, the corresponding improvements were 50 and 80 percent.

4.1.3 Emission Control Devices

4.1.3.1 Exhaust Gas Recirculation

4.1.3.1.1 General

Tests conducted by a number of investigators have indicated that incorporation of exhaust gas recirculation is an effective means to lower the NO_x emissions from open-chamber as well as divided-chamber diesel engines. Generally, this reduction is associated with some loss in engine performance and an increase in smoke.

The mechanism involved in reducing NO_x is related to the intake charge dilution obtained with EGR, resulting in lower combustion temperatures and lower oxygen concentrations. Both effects have a tendency to inhibit the formation of NO_x during the combustion process, particularly at the lower air-fuel ratios associated with full-load operation. Best results, in terms of NO_x reduction and associated engine performance loss, have been achieved when the exhaust gases were cooled before recycling. Tests with hot EGR have resulted in substantially higher fuel consumption and smoke emissions.

4.1.3.1.2 Experimental Programs

The effects of EGR on the NO_x and smoke emissions of a naturally aspirated diesel engine are illustrated in Figure 4-17 (Ref. 4-4). In these tests, the EGR flow was cooled to the temperature of the intake air. As indicated in the figure, substantial reductions in NO_x were achieved with moderate EGR flow rates. For instance, utilization of ten percent EGR resulted in a reduction in NO_x of this

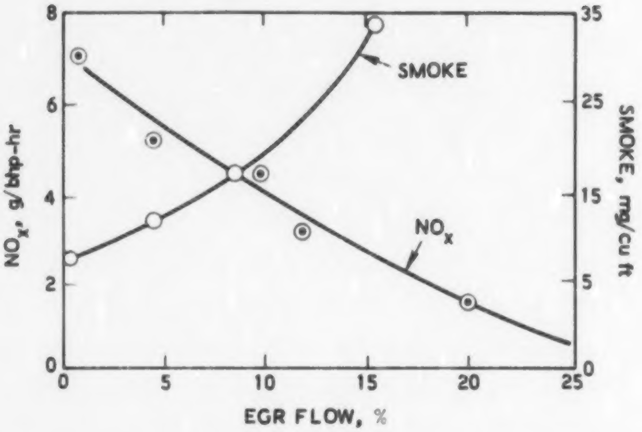


Figure 4-17. Effect of EGR on NO$_x$ and smoke emissions of a naturally aspirated open-chamber diesel engine (Ref. 4-4)

engine from about 7 g/bhp-hr to about 4 g/bhp-hr. However, the smoke density (soot) of the engine exhaust increased in this case from about 8 mg/ft^3 to about 22 mg/ft^3. The increase in smoke is the direct result of a reduction in oxygen availability in the combustion chamber, which also is responsible for the observed loss in engine power output capability and the increase in CO emissions. These problems can be alleviated by adding a turbocharger.

Pischinger (Ref. 4-7) conducted EGR tests on a 610 CID multicylinder, four-stroke, open-chamber, low-swirl diesel engine which was operated at an intermediate speed of 1400 rpm and at part-load conditions (52 psi BMEP). NO$_x$ decreased almost linearly with increasing EGR flow rate, at the expense of some increase in smoke and fuel consumption. HC was rather independent of EGR, while CO showed no change up to 30 percent EGR, but increased rapidly for higher EGR rates. The use of 10 percent EGR resulted in a 20 percent reduction in NO$_x$ at practically zero loss in fuel consumption. With 20 percent EGR, NO$_x$ was diminished by about 30 percent, while the specific fuel consumption suffered a loss of two percent. No information is available regarding the temperature of the EGR flow used in these tests. Since the reported effect of EGR was less than observed by other investigators using cooled exhaust, it is conceivable that uncooled EGR was employed in this particular investigation.

Emission versus EGR curves computed from the data published by the Bureau of Mines (Ref. 4-3) for a naturally aspirated, open-chamber diesel (engine No. 7 in Table 3-2) are presented in Figure 4-18. In this case, the EGR flow was extracted near the muffler and was cooled before induction into the intake manifold of the engine. As indicated, the NO_x emissions decreased rapidly with increasing EGR, particularly at low EGR flows. As expected, EGR was more effective under high load conditions because of the higher combustion temperatures prevailing at the higher load points and the lower oxygen concentration in the exhaust gases. Also, EGR appears to be more effective at higher engine speeds. As indicated, utilization of ten percent EGR resulted in a reduction in NO_x of about 60 percent at full load and 45 percent at half load. Under these conditions, the specific fuel consumption increased by about three percent and the peak smoke intensity increased by about 80 percent. At half load, the CO emissions remained essentially constant, whereas a substantial increase in CO was observed at full load for EGR flow rates above about eight percent. The HC emissions show some reduction with increasing EGR flow rate, particularly under full-load operating conditions of the engine. Conversely, the peak smoke intensity increased markedly with EGR, while the specific fuel consumption showed only a small increase. The power output capability of the engine deteriorated mildly with increasing EGR flow.

EGR test data from a four-stroke, turbocharged, open-chamber engine rated at 230 bhp are presented in Figure 4-19 (Ref. 4-2). As shown, ten percent EGR resulted in a 35-percent reduction of NO_x.

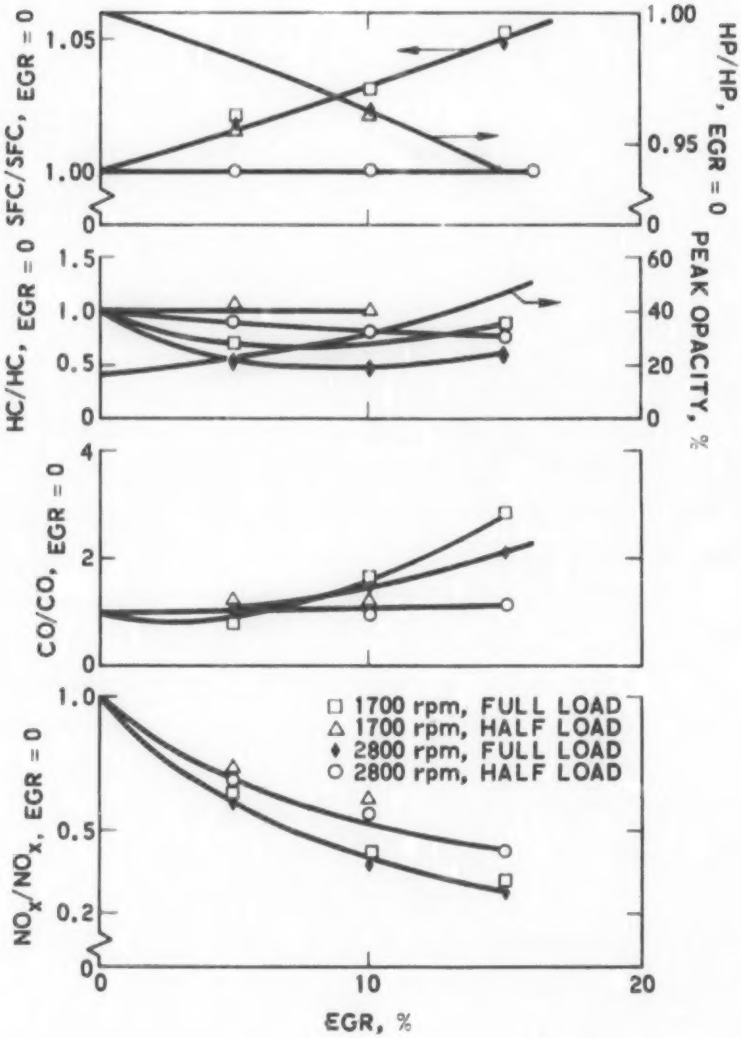


Figure 4-18. Effect of EGR on naturally aspirated open-chamber diesel engine mass emission and performance, Engine No. 7 (Ref. 4-3)

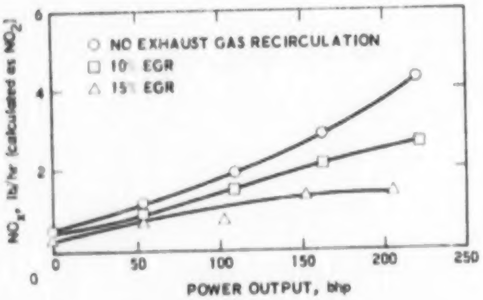


Figure 4-19. Effect of EGR on NO_x emissions of a turbocharged, open-chamber diesel engine at rated speed (Ref. 4-2)

Apparently, there was little effect on the fuel consumption and power output capability of the engine. Further reduction in NO_x was achieved with higher EGR rates, but the performance of the engine suffered greatly, particularly in terms of low-speed power capability. According to Figure 4-19, EGR is more effective at the higher power levels because the oxygen content in the recirculated gas decreases with increasing load.

NO_x emission data from a divided-chamber, turbocharged diesel engine indicate a nearly linear reduction of NO_x with increasing EGR flow rate. With ten percent EGR, NO_x was reduced by about 40 percent, and with 15 percent EGR, the average reduction was about 65 percent (Ref. 4-1).

The effectiveness of EGR in an air-scavenged, two-stroke diesel engine, rated at 148 hp and 2100 rpm, is shown in Figure 4-20. At 80 hp output, the use of 25 and 40 percent EGR resulted in NO_x reductions of about 70 percent and 90 percent, respectively. However, the engine had to be derated by as much as 40 percent to stay within the smoke limits of the non-modified engine. Also, HC and CO emissions increased substantially with this high EGR rate and this required incorporation of a catalytic converter in the exhaust system (Ref. 4-2).

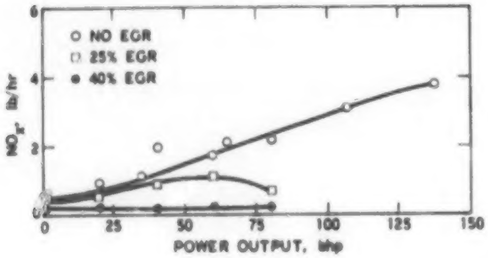


Figure 4-20. Effect of exhaust gas recirculation on NO_x emission of an air-scavenged, two-stroke diesel, at rated speed (Ref. 4-2)

Single-cylinder, naturally aspirated diesel engine data are presented in Figure 4-21 (Ref. 4-23). In these tests, the EGR flow was cooled to 80° F before induction into the engine intake, and the fuel flow rate was held constant, resulting in an air-fuel ratio of 24.8 at zero EGR and 19.9 at 20 percent EGR. Again, NO_x decreased approximately linearly with increasing EGR flow rate, whereas smoke shows an inverse relationship. Up to about 12 percent EGR, the specific fuel consumption of the engine remained constant, and HC was not affected at all by the use of EGR. Utilization of ten percent cooled EGR resulted in a 40-percent reduction in NO_x.

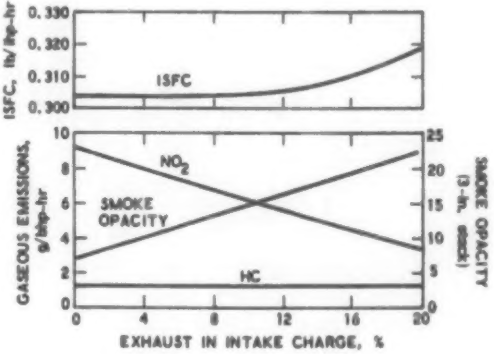


Figure 4-21. Effect of EGR on performance and emissions of a single-cylinder naturally aspirated engine (80 psi bmep, cooled EGR at 80°F) (Ref. 4-23)

Experimental results from a single-cylinder, open-chamber diesel engine operated at 2100 rpm and 143 psi BMEP, indicate a 25 percent reduction in NO_x when using 10 percent EGR (Ref. 4-8). With 20 percent EGR, the observed NO_x reduction was about 45 percent.

Exhaust gas recirculation tests over the 13-mode cycle were conducted by Ricardo and Company, using an 855 CID, divided-chamber diesel engine (Ref. 4-24). In these tests, the EGR flow was cooled to 122°F, and the rate of EGR admission was varied over the cycle, employing 20 percent EGR at loads up to 50 percent, ten percent EGR at 75 percent, and zero EGR at full load. Under these conditions, the NO_x emissions over the cycle were reduced from 7.1 g/bhp-hr to 5.3 g/bhp-hr, an improvement of about 25 percent.

One manufacturer of large stationary diesel engines has reported that a 45-percent reduction in NO_x was achieved in an exploratory test program, using 15 percent EGR. In these tests, the specific fuel consumption showed no change with EGR, but the smoke intensity increased by about 100 percent. The observed NO_x reduction is in reasonable agreement with the test data discussed above. Another manufacturer feels that a 20-percent reduction in NO_x might be achieved without a loss in specific fuel consumption. However, a reduction in NO_x by 40 percent would be accompanied by a 10-percent loss in specific fuel consumption.

4.1.3.1.3 Potential Problem Areas

Although basically feasible, most manufacturers of heavy-duty diesel engines feel that a number of potential problem areas would have to be resolved before incorporation of EGR into stationary engines could become a reality. These problems include corrosion and deposit buildup in the EGR circuit, and in the case of turbocharged engines, fouling of the compressor and intercooler. Furthermore, the long-term effect of EGR on engine wear and lube oil contamination is currently not known, and systematic tests would be required to determine the magnitude of these effects. In any case, one manufacturer feels that engine maintenance costs would definitely increase with the application of EGR, and another expressed concern that the problems with EGR in diesels might be more severe than in gasoline engines because of the higher sulfur content in diesel fuel relative to gasoline.

4.1.3.1.4 Projected EGR Characteristics

The available test data indicate significant variations in the observed effects of EGR on the emission and performance characteristics of diesel engines. This is illustrated in Figure 4-22, showing data bands containing all the available test data discussed in Section 4.1.3.1.2. When using these curves, consideration must be given to the fact that the effects of EGR depend on many factors including engine type, operating speed and load, and EGR temperature. Currently, there are insufficient data available to derive accurate correlations for all these parameters. Cummins has developed a fairly good correlation for medium speed diesels, using the oxygen concentration in the intake manifold as the correlating parameter (Ref. 4-14).

Based on the data presented here, it is concluded that EGR is an effective technique to reduce NO_x emissions from diesel engines. However, the full benefits of EGR cannot be realized without some sacrifice in engine performance and some increase in CO and smoke emissions.

4.1.3.2 Water Addition

4.1.3.2.1 General

Like EGR, water added to the intake air or injected into the cylinder of diesel engines, either separately or in the form of water-fuel emulsions, acts as a charge diluent, resulting in lower compression

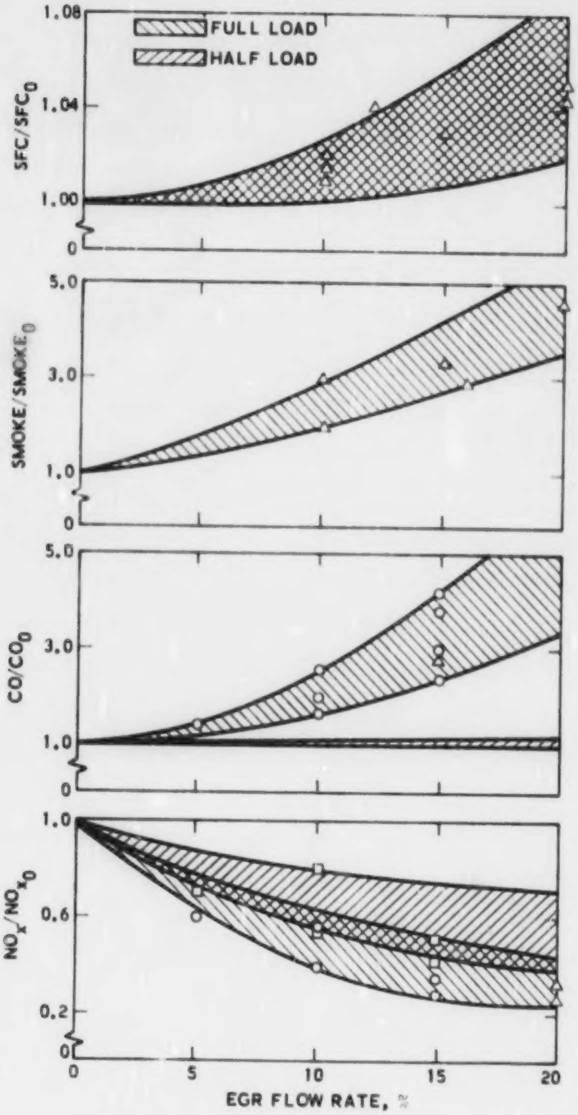


Figure 4-22. Effect of EGR on diesel engine emissions and specific fuel consumption

and combustion temperatures and oxygen concentrations, hence lower NO_x formation rates.

Water inducted with the intake air is superheated during the compression stroke of the piston and distributed fairly homogeneously throughout the combustion chamber. In this case, the temperature at the start of combustion is not significantly different from the no-water case, since the mass of water inducted is generally small, relative to the mass of air. As a result, the ignition characteristic of the charge does not change much relative to operation without water.

Conversely, when water is injected directly into the engine cylinder as part of a fuel-water emulsion, vaporization of the water and mixing with the air takes place locally. Since water has a high latent heat, vaporization of the water causes a significant reduction in the local temperature, resulting in an increase in the ignition delay. Therefore, to achieve satisfactory operation of the engine, the fuel injection timing must then be advanced, which negates the expected benefits of the water (Refs. 4-4 and 4-9).

4.1.3.2.2 Water Induction

The effect of water induction on diesel engine emissions has been studied by a number of investigators using open-chamber and divided-chamber, naturally-aspirated, and turbocharged engines.

Results obtained by Cummins (Ref. 4-9) on a single-cylinder, open-chamber diesel engine are presented in Figure 4-23, showing NO and HC concentration and smoke measurements as a function of the water flow rate for two different engine speeds. Also shown in this figure are data obtained with emulsified fuel, which are further discussed in Section 4.1.3.2.3. As indicated, the NO_x and smoke emissions decreased markedly with increasing water flow rates, and the observed effects were nearly independent of engine speed. For example, for a water flow rate of 40 percent (by volume) of the combined water plus fuel flow, NO_x was reduced by about 25 percent and

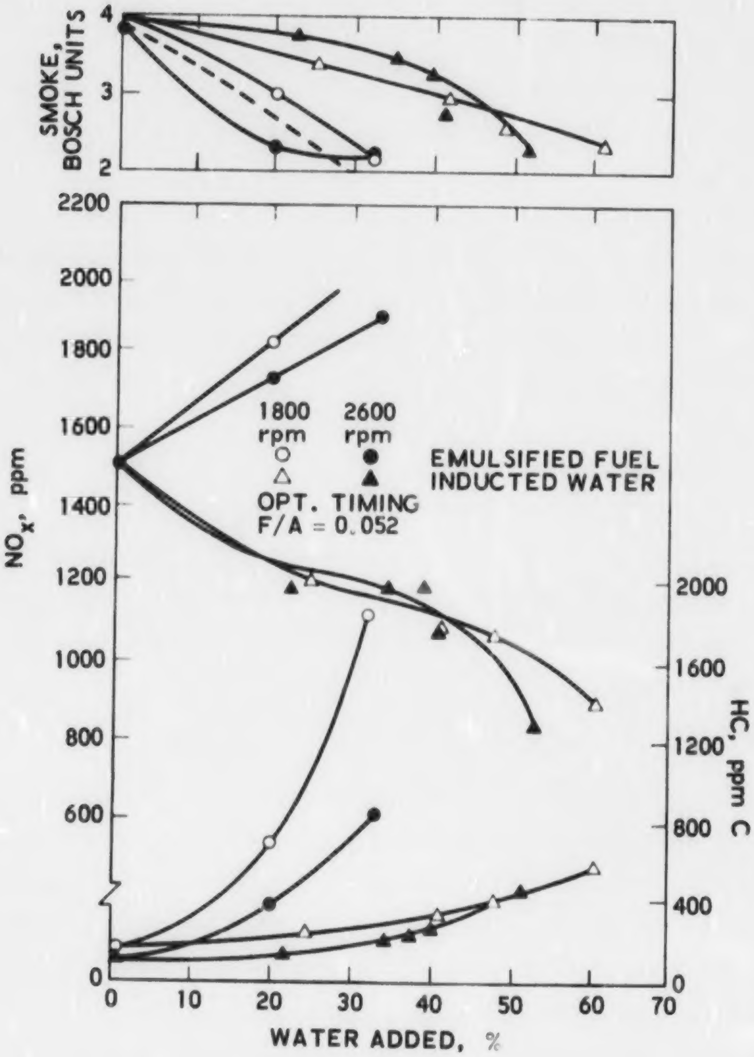


Figure 4-23. Effect of inducted and emulsified water on the HC, smoke, and NO$_x$ emissions of an open-chamber diesel engine (Ref. 4-9)

the smoke level was reduced by about 20 percent. However, under these conditions, HC increased by about 150 percent, whereas little change was observed in CO. The indicated mean effective pressure decreased slightly with increasing water flow rate. It should be noted that the air-fuel ratio was held constant during these tests and the injection timing was set at the optimum value for each test point.

Tests with water induction, conducted by Ricardo and Company (Ref. 4-24) on a multicylinder, open-chamber diesel engine, resulted in substantial NO_x reductions, with a concomitant small increase in CO and HC, and a slight loss in power. In these tests, fuel injection timing was set for optimum performance and the engine was operated over the 13-mode test cycle. Emission measurements were performed for water-to-fuel-flow ratios of 0.5 and 1.0. The test data, listed in Table 4-4, indicate that a 50-percent reduction in NO_x was achieved for a water-to-fuel-flow ratio of 1.0. Again, the NO_x reduction was approximately proportional to the amount of water added.

The effect of water injection into the intake system of a four-stroke, turbocharged, open-chamber diesel engine is illustrated in Figure 4-24, showing NO_x concentration as a function of water vapor content in the intake air (Ref. 4-2). As indicated, the NO_x reduction

TABLE 4-4. EFFECT OF WATER INDUCTION ON THE EMISSIONS OF AN OPEN-CHAMBER DIESEL ENGINE (13-MODE CYCLE DATA)

Water/Fuel Ratio	Emissions, g/bhp-hr			Power Loss, %	NO_x Reduction, %
	CO	HC	NO_x		
0.1	3.5	0.5	15.5	-	-
0.5	4.3	0.5	11.4	2	26
1.0	4.7	0.6	7.8	4	50

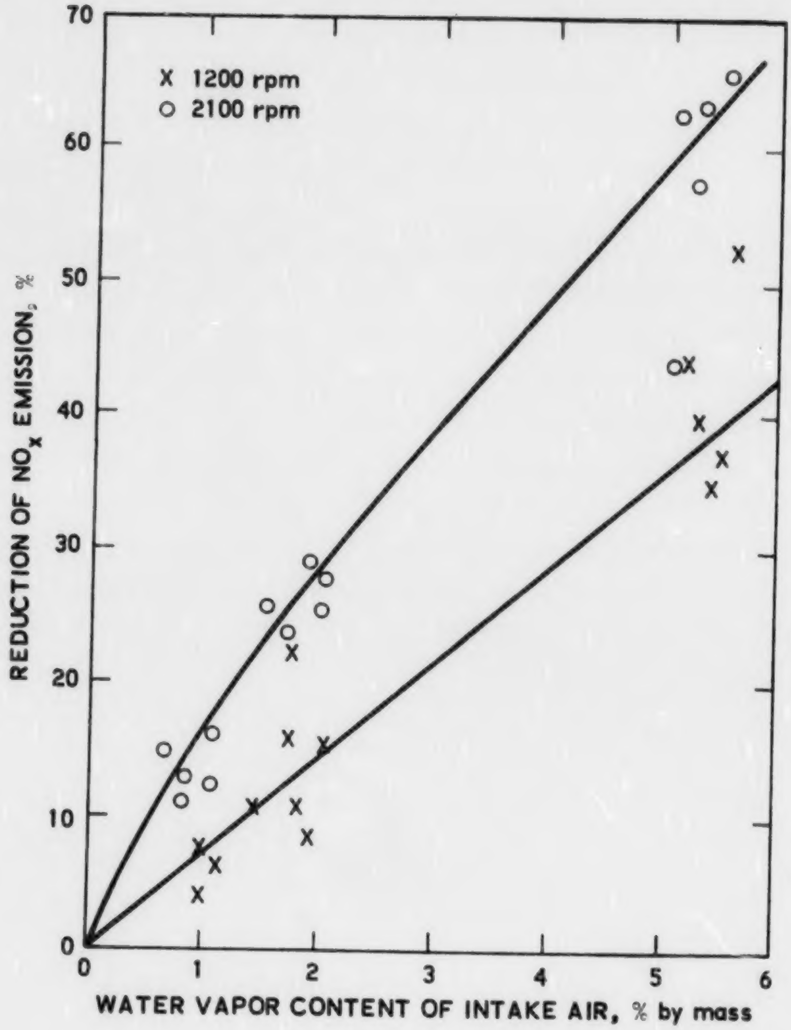


Figure 4-24. Reduction of NO$_x$ emission as a function of water induction — turbocharged, open-chamber diesel engine (Ref. 4-2)

observed on this engine is approximately proportional to the water flow rate. Utilization of four percent water (by mass) resulted in a NO_x reduction of about 50 percent at 2100 rpm and about 30 percent at 1200 rpm. Under full-load conditions, the water flow rate used in these tests was approximately equal to the fuel flow rate. Apparently, water addition resulted in no significant detrimental effects on CO, HC, and specific fuel consumption.

Figure 4-25 presents NO_x emission data for a turbocharged, divided-chamber engine which was operated at maximum torque and 60-percent torque, using water-to-fuel-flow ratios between 0 and 2.5 (Ref. 4-1). For a water-to-fuel-flow ratio of 1.0, the observed reduction in NO_x was about 70 percent at the high torque condition and about 65 percent at the intermediate torque point. As shown in the figure, water induction is particularly effective for water-to-fuel-flow ratios up to about 1.0. Beyond that point, the gains diminish, while the probability of water condensation in the intake system increases. In these tests, the water was added ahead of the turbocharger. Subsequent analyses conducted by Caterpillar indicate,

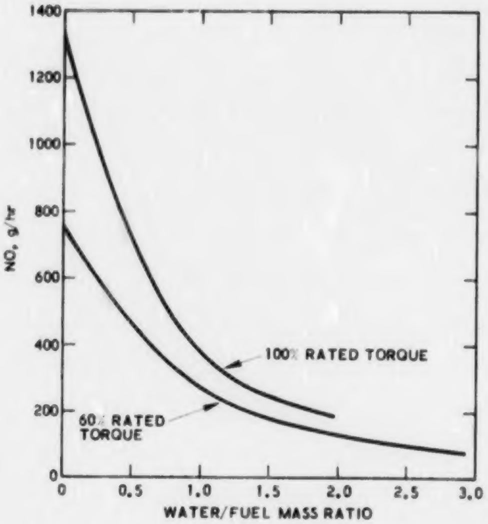


Figure 4-25. Water induction versus NO emission — prechamber, turbocharged diesel, 2200 rpm (Ref. 4-1)

however, that adding the water just ahead of the intake valve or directly into the cylinder itself would alleviate potential problems related to water condensation and "settling out" of the water in the intake system (Ref. 4-1). Steam injection into the engine has been considered as an alternate method which might provide a better and more effective distribution of the water throughout the combustion chamber. However, the heat of vaporization of the water would not be available in this case and this would tend to lower the effectiveness of this technique.

The effects of water induction on NO_x, soot, and indicated specific fuel consumption of a single-cylinder, open-chamber and a divided-chamber diesel engine were determined by Wilson et al. (Ref. 4-5). Again, NO_x decreased with increasing water flow rate, particularly for water-to-fuel-flow rates up to about 1.0, while SFC showed very little change. However, with increasing water induction, soot formation showed a tendency to increase somewhat. For both engine types, operation at water-to-fuel-flow rates of 0.5 and 1.0 resulted in NO_x reductions of 35 percent and 60 percent, respectively. The data indicate some variations with engine speed and air-fuel ratio (load), but the observed trends are not consistent.

Water induction data reported by Cooper-Bessemer for a large stationary diesel engine are listed in Table 4-5 (Ref. 4-6). In these tests, the water was injected into the intake air manifold of each cylinder and the engine was operated at rated speed and load conditions, using either No. 2 diesel fuel or natural gas. In the tests with diesel fuel, a water flow rate of 0.5 gallons per minute was used, which corresponds to a water-to-fuel-flow rate of about 0.17. This resulted in a seven-percent reduction in NO_x. Assuming a linear relationship between NO_x reduction and water addition, a 40-percent reduction in NO_x is predicted for this engine when operated at a water-to-fuel-flow ratio of about 1.0, compared with about 50 to 60 percent for the

TABLE 4-5. EFFECT OF WATER INDUCTION ON COOPER BESSEMER KSV-12 DIESEL ENGINE EMISSIONS AND FUEL CONSUMPTION

Fuel	rpm	Load	Water Rate, gpm	Mass Emissions, g/bhp-hr			BSFC,[a] $\frac{Btu}{hp\text{-}hr}$	$\frac{HC^b}{HC_0}$	$\frac{CO}{CO_0}$	$\frac{NO}{NO_{x_0}}$	$\frac{BSFC^a}{BSFC_0}$
				HC	CO	NO_x					
No. 2 Diesel	600	Full	0	0.13	3.85	10.99	6677	-	-	-	-
No. 2 Diesel	600	Full	0.5	0.16	4.25	10.20	6643	1.23	1.10	0.93	0.995
Natural Gas	600	Full	0	5.16	4.50	8.96	6340	-	-	-	-
Natural Gas	600	Full	0.3	6.48	3.39	8.35	6374	1.26	0.75	0.93	1.005

[a] Brake Specific Fuel Consumption

[b] Subscript zero refers to operation without water

smaller, higher-speed engines discussed above. The data listed in Table 4-5 show relatively small variations in HC, CO, and SFC, and this might be due to data scatter.

4.1.3.2.3 Emulsions

Single-cylinder engine tests were conducted by Cummins to determine the effect of emulsified fuel on diesel engine emissions (Ref. 4-9). These tests showed that the emulsified fuel had a marked effect on the combustion process, and ignition timing advance was required to compensate for the increasingly larger ignition delays obtained with increasing water content in the emulsion. As a result, the NO_x emissions increased with increasing water content, as shown in Figure 4-23. Also, the HC emissions increased substantially with increasing water content.

4.1.3.2.4 Projected Characteristics

The available data on water injection into the intake system of diesel engines are summarized in Figure 4-26, showing the percentage reduction in NO_x as a function of the water-to-fuel mass ratio. As indicated, there is a large engine-to-engine variability in the data which might be due to variations in certain engine design and operating parameters. However, in most cases, there is insufficient information available to establish correlations between the NO_x reduction and water flow rate for the different engine categories. Moreover, there is a lack of data relating water induction rate and HC, CO, smoke, and specific fuel consumption.

4.1.3.2.5 Potential Problem Areas

Although water injection appears to be an effective technique to reduce the NO_x emissions from diesel engines, most manufacturers are concerned about potential problem areas directly related to the use of water. These include corrosion and wear of intake system

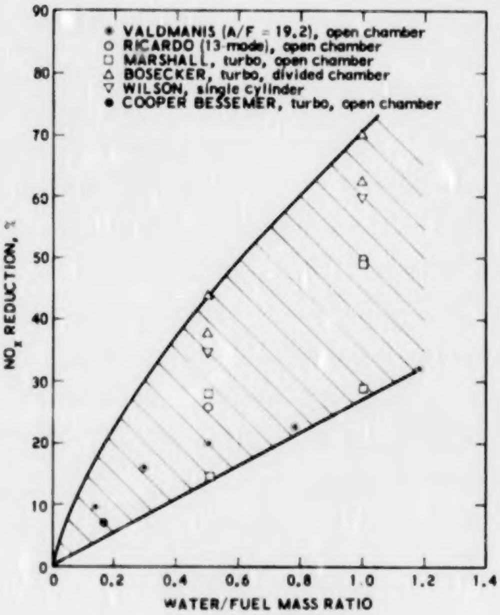


Figure 4-26. NO_x reduction versus water to fuel mass ratio

components, such as valves and manifolds, as well as fouling of the water injection nozzles. To minimize corrosion and deposit buildup, one manufacturer suggests the use of distilled water. Other difficulties include the requirement of a water flow control system to meter the flow in proportion to the intake air flow rate, and a water pressurization system to overcome the high intake manifold pressure in turbocharged engines. Also, the water must be protected from freezing, preferably without the addition of alcohol, which would probably increase the HC emissions of the engine (Ref. 4-9). Since a portion of the inducted water may find its way into the engine crankcase, more frequent oil changes might be required if water injection would be incorporated. Currently, there is insufficient information available to allow a comprehensive assessment of the long-term effects of water injection on the performanc[illegible]nd durability characteristics of diesel engines.

4.1.3.3 Catalytic Converters

4.1.3.3.1 General

During the past few years, considerable efforts have been directed by the automotive industry and other organizations toward the development of catalytic converters for the control of exhaust emissions from spark ignition engine powered automobiles. Although catalytic performance degradation with mileage accumulation has been a serious problem area in the past, a number of catalytic systems have now reached an advanced state of development and have been shown to be quite effective in reducing the HC and CO exhaust emission levels from automobiles. However, progress has been rather slow in the development of practical NO_x reduction catalysts. The NO_x catalysts currently considered by the automobile manufacturers are dependent upon the availability of adequate amounts of reducing species in the engine exhaust, such as CO or H_2 to convert NO to $N_2 + CO_2$ or to $N_2 + H_2O$. Since CO is practically nonexistent in diesel exhaust, the currently known reduction catalysts are not considered to be effective for diesels.

It has been suggested to produce the required amounts of CO and/or H_2 by means of a methane or propane burner installed in the exhaust system or by adding ammonia to the exhaust gases (Ref. 4-1). Although theoretically feasible, there is insufficient data available at this time to permit a meaningful assessment of these approaches. In any event, the added engine operating cost and complexity resulting from the use of these techniques and the requirement of subsequent oxidation of any excess CO in the exhaust would have to be weighed against other potential NO_x abatement methods.

4.1.3.3.2 Experimental Results

To date, a limited amount of exploratory catalyst test work has been performed on diesel engines. For example, Cummins has run a commercially available catalyst, which was specifically designed for diesel applications, in one of its naturally aspirated, open-chamber diesel engines (Ref. 4-4). In these tests, CO was reduced by about 90 percent and HC was reduced by about 50 percent, while no change was observed in NO_x.

The HC and CO reductions reported by Caterpillar (Ref. 4-1) for a catalyst installed in the exhaust system of a prechamber diesel engine are presented in Figure 4-27. At full load, the HC and CO conversion efficiencies are over 90 and 80 percent, respectively. However, with decreasing load, the effectiveness of the catalyst decreases as a result of the lower exhaust gas temperatures and the associated reduction in the HC and CO oxidation reaction rates. Also shown in Figure 4-27 are test data obtained by the Bureau of Mines (Ref. 4-3) from a naturally aspirated, open-chamber diesel engine which was fitted with two Universal Oil Products (UOP) catalysts. As indicated, the trends in HC and CO reduction versus engine load are in reasonable agreement with the Caterpillar data, but the magnitude of

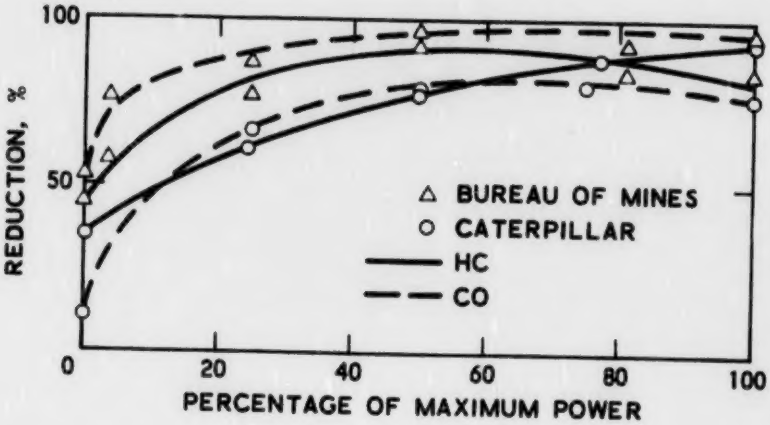


Figure 4-27. Effect of catalysts on diesel engine emissions

the observed reductions are higher in the case of the UOP catalysts. According to Ref. 4-3, incorporation of the catalysts resulted in a substantial reduction in aldehyde emissions and exhaust odor intensity while the smoke level remained essentially unchanged. With catalysts, the engine back-pressure increased by a factor of 2, relative to the standard configuration, and this caused a small loss in the power output capability of the engine.

Similar results were obtained by the Bureau of Mines on another naturally aspirated, open-chamber engine (Engine No. 7 in Table 3-2), equipped with two Engelhard noble metal catalysts, and by Pischinger and Cartellieri (Ref. 4-7). Again, incorporation of the catalysts had no effect on NO_x and smoke, but resulted in some reduction of the exhaust odor intensity.

A number of base metal and noble metal catalysts were tested by Southwest Research Institute in a city bus which was operated over its normal duty cycle (Ref. 4-25). Some of the catalysts were experimental and some were "off the shelf" but intended for different applications. In several cases, installation of the catalyst resulted in lower emissions of HC, CO, odor, and certain partially oxygenated materials. No beneficial effect on smoke was detected in any of these tests.

It should be noted that the results presented above for a number of catalyst installations were achieved with fresh catalysts and do not include potential catalyst degradation effects due to accumulation of test time.

4.1.3.4 Thermal Reactors

A thermal reactor is a chamber replacing the conventional exhaust manifold system of the engine and is sized and configured to increase the residence time of the exhaust gases and permit further chemical reactions, thus lowering the HC and CO emissions. Devices of this type have been successfully applied in automotive spark ignition engines, but are not considered to be very effective in diesel engines. At most operating conditions, the temperature of the diesel exhaust and the concentration of HC and CO are believed to be too low to sustain the HC and CO oxidation reactions. More importantly, NO_x, which represents the major pollutant specie in diesels, is not substantially altered in thermal reactors.

4.1.3.5 Exhaust Scrubbers

Exhaust scrubbers, which have been used in mining vehicles, are not considered feasible for application in stationary diesel engines. Testing of scrubbers conducted by one engine manufacturer indicates some reduction in HC and particulate emissions. However, the most toxic species – CO and NO_x – remained unchanged in these tests.

4.1.3.6 Turbocharging

The power output of naturally aspirated diesels is frequently limited by excessive amounts of smoke which is formed when operating at low air-fuel ratios. Incorporation of a turbocharger increases the air flow rate of the engine and permits operation at higher air-fuel ratios. As a result, the smoke-limited power cutoff point of the engine moves to higher levels. With turbocharging, the manifold temperature increases, causing higher combustion temperatures and NO_x formation rates. These trends can be counteracted to some degree by means of an intercooler installed between the compressor of the turbocharger and the engine intake manifold.

The effect of turbocharging on a family of 404 CID four-stroke medium-swirl, open-chamber diesel engines is illustrated in Figure 4-28, showing NO_x, brake specific fuel consumption, and smoke levels at rated speed as a function of brake mean effective pressure (power output) (Ref. 4-23). Incorporation of a turbocharger increases

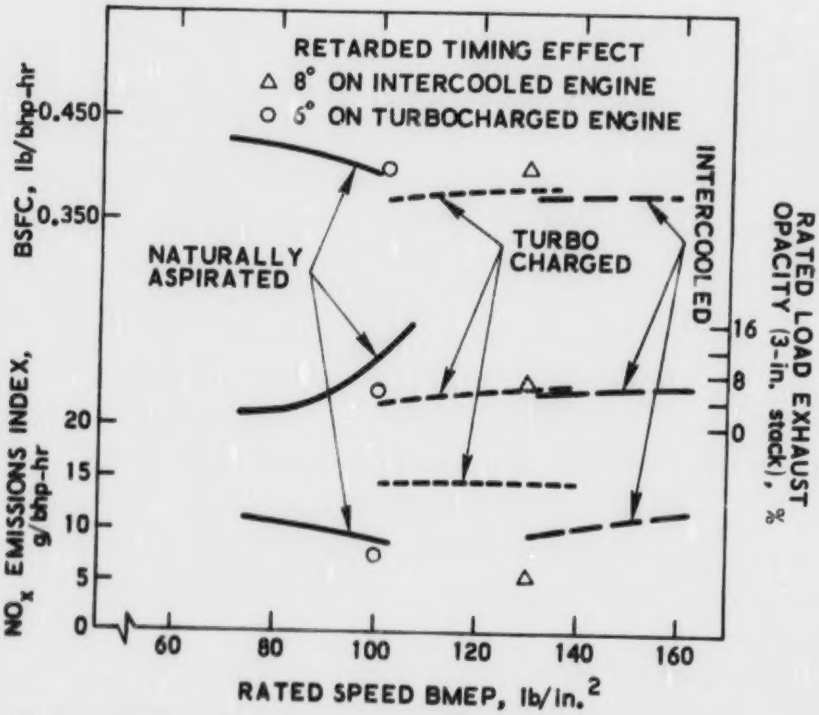


Figure 4-28. Effect of turbocharging and intercooling on the emissions and specific fuel consumption of a family of open-chamber diesel engines (Ref. 4-23)

the power output capability of the engine and results in better specific fuel consumption and lower exhaust smoke levels. With an intercooler, NO_x can be further reduced to the levels of the naturally aspirated engine while maintaining lower SFC and exhaust smoke. As shown in Figure 4-28, retarding the injection timing of the turbocharged engine by six degrees increased the specific fuel consumption to the level of the naturally aspirated engine, but reduced NO_x from about 12 g/bhp to 7 g/bhp-hr. This represents a reduction of about 12 percent relative to the naturally aspirated engine. With intercooling and eight degrees retard, the NO_x emissions were further reduced to about

5.3 g/bhp-hr, while maintaining the specific fuel consumption of the naturally aspirated engine. This corresponds to a 35-percent reduction in NO_x relative to the naturally aspirated engine. In general, timing retard tends to increase the smoke emissions, but the rate of change observed on this engine family was rather small.

Similar results are presented in Figure 4-29, showing curves computed from test data published by the Bureau of Mines (Engine No. 8 in Table 3-2) (Ref. 4-3). The emission and SFC values plotted are referenced to the corresponding values obtained without turbocharging (subscript 0). As indicated, turbocharging results in a reduction in SFC and CO, particularly in the high power regime, but causes higher emissions of NO_x and HC. The effect of engine speed on SFC, HC, and CO is negligible, but speed does affect NO_x to some degree. Although not shown in the figure, the addition of the turbocharger resulted in a reduction in the peak smoke opacity of the engine from 20 to about 2 percent.

Tests conducted by Pischinger on open-chamber and prechamber engines (Ref. 4-7) indicate a 20-percent increase in NO_x with turbocharging. However, incorporation of an aftercooler brought the NO_x emissions of the turbocharged engine back to the level obtained without turbocharging.

Based on these data, it is concluded that incorporation of a turbocharger and intercooler, combined with injection timing retard, represents an effective technique to reduce smoke and NO_x emissions without incurring a loss in specific fuel consumption relative to operation without a turbocharger.

4.1.4 Emission Control Systems

The emission and fuel consumption characteristics of several combinations of the emission control devices/techniques described in Sections 4.1.1 through 4.1.3 were evaluated by a number

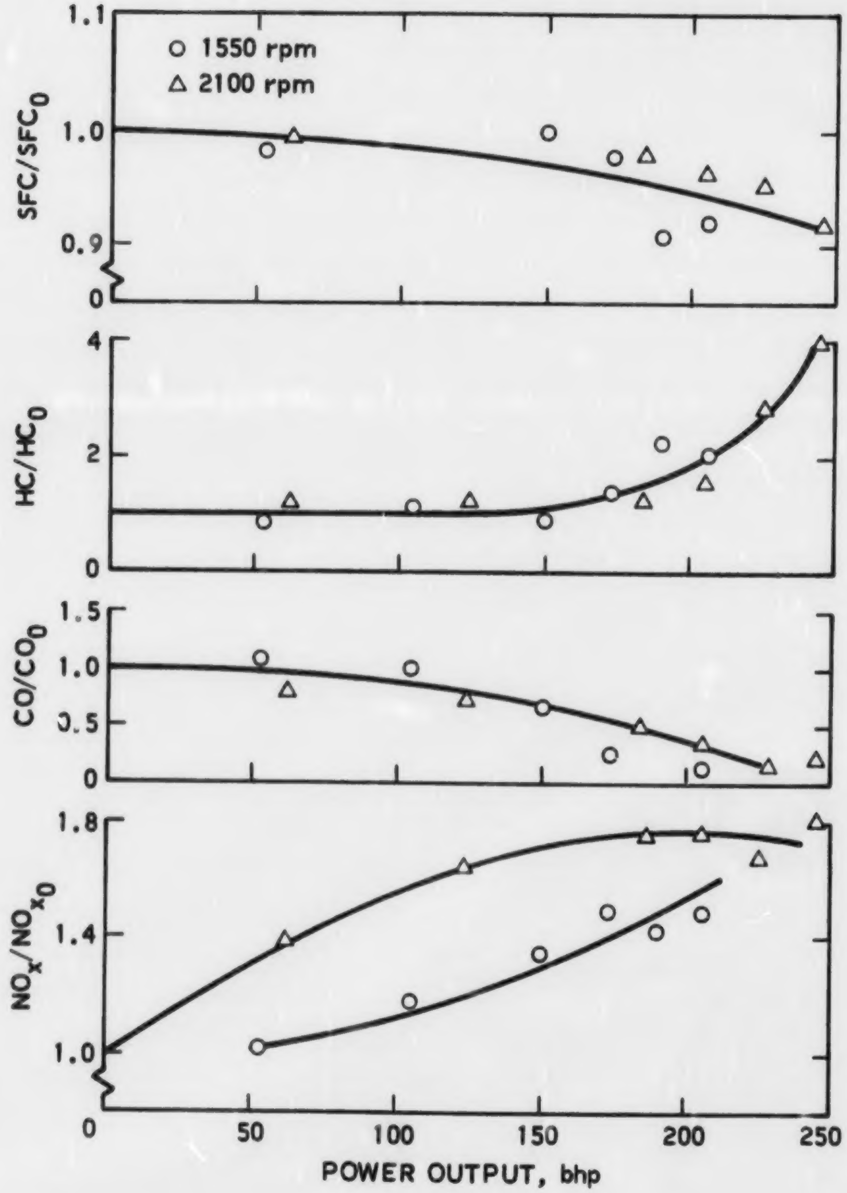


Figure 4-29. Effect of turbocharging on specific fuel consumption and mass emissions (Engine No. 8) (Ref. 4-3)

of organizations including the Bureau of Mines (Refs. 4-2 and 4-3), Cooper-Bessemer (Ref. 4-6), and Cummins Engine Company (Ref. 4-26). The majority of the Bureau of Mines tests was conducted over the 13-mode, heavy-duty engine cycle, with the remainder consisting of constant-speed data covering a wide range of engine load conditions. The test data reported by Cooper-Bessemer are from one of its large production engines operated at rated speed and load on both No. 2 diesel fuel and natural gas. The Cummins data represent 13-mode projections for three different emission control systems applied to present production engines. The data are briefly discussed in the following sections.

4.1.4.1 Thirteen-mode Data

Specific mass emissions of HC, CO, and NO_x from a 230-bhp, four-stroke, turbocharged, open-chamber diesel engine operated over the 13-mode cycle are listed in Table 4-6. The engine was equipped with several emission control systems consisting of combinations of intake air cooling, EGR, and water injection. Minimum NO_x mass emissions of 6.4 g/bhp-hr (52-percent reduction) were achieved on this engine with 15 percent EGR plus intake air cooling and with water induction at a rate of six percent (by mass of total intake flow) plus intake air cooling. The variations in CO and HC obtained in these tests follow the trends observed with the individual control techniques discussed in Sections 4.1.1 through 4.1.3.

Table 4-7 summarizes the emission and specific fuel consumption results obtained by the Bureau of Mines for a number of production engines (Ref. 4-3). These engines, identified in Tables 3-2 through 3-7, were operated over the 13-mode cycle and were equipped with experimental emission control systems consisting of various combinations of injection timing modifications, EGR, intake cooling, modified injectors, and oxidation catalysts.

TABLE 4-6. EFFECT OF EMISSION CONTROL SYSTEMS ON THE EMISSIONS OF A TURBOCHARGED, OPEN-CHAMBER DIESEL ENGINE – 13 MODE CYCLE (Ref. 4-2)

Configuration	Carbon Monoxide, g/bhp-hr	Hydrocarbon Calculated as CH_2, g/bhp-hr	Nitrogen Oxides Calculated as NO_2, g/bhp-hr	Percentage Change[a]		
				CO	HC	NO_x
Standard	4.5	1.9	13.3	-	-	-
Intake Air Aftercooled						
50°F	3.4	2.0	9.9	-25	+5	-25
185°F	3.1	1.8	12.1	-31	-5	-9
Intake Air Aftercooled and Cooled Exhaust EGR (185°F)						
10% EGR rate	3.8	1.2	8.7	-15	-37	-35
15% EGR rate	6.3	1.2	6.4	+40	-37	-52
Intake Air Aftercooled (185°F) and Water Injection						
1% water by mass	2.2	2.0	11.5	-51	+5	-13
2% water by mass	2.3	2.1	10.6	-49	+10	-20
6% water by mass	3.1	1.4	6.4	-31	-26	-52

[a] Referenced to standard engine

Results from Engine No. 7 show that NO_x can be reduced from the 8.3 g/bhp-hr level of the standard engine to about 4.2 g/bhp-hr by combining three degrees retarded injection timing and ten percent EGR. This corresponds to a 50 percent reduction in NO_x. However, a seven percent loss in fuel consumption was incurred under these conditions. Using only five percent EGR resulted in a 40 percent

TABLE 4-7. SUMMARY OF EMISSIONS AND FUEL CONSUMPTION FOR BASELINE AND COMBINATION OF PARAMETERS TESTS FOR DIESEL ENGINES

Control System	Description of Engine Adjustment and Accessory Hardware	13-Mode Cycle						Lug-Down Smoke, % Opacity	Percentage Change[c]						
		Emissions, g/bhp-hr			BSFC,[a] lb/bhp-hr	Aldehydes, g/bhp-hr	Odor Intensity, DI Units[b]		CO	HC	NO_x	BSFC[a]	Aldehydes	Odor	Smoke
		CO	HC	NO_2											
	Baseline, Engine No. 7[d]	5.93	2.90	8.27	0.445	0.23	4.3	13.3	-	-	-	-	-	-	-
1	Standard Timing, 10% EGR, Englehard Catalyst, (EGR cut off at full load)	1.73	1.22	6.24	0.454	0.16	3.6	16.6	-71	-58	-25	+2.0	-30	-12	+25
2	3° Retarded Timing, 5% EGR, Englehard Catalysts, (EGR cut off at full load)	1.89	1.78	4.97	0.453	0.29	3.8	17.5	-68	-39	-40	+1.8	+26	-7	+31
3	3° Retarded Timing, 10% EGR, Englehard Catalysts, (EGR cut off at full load)	1.93	1.79	4.18	0.476	0.20	4.0	14.0	-67	-38	-49	+7.0	-13	-2	+5
	Baseline, Engine No. 24[d]	0.83	0.26	4.93	0.422	0.082	4.6	6.9	-	-	-	-	-	-	-
1	Standard Timing, 10% EGR, No Aftercooling	1.20	0.28	3.13	0.421	0.028	1.9	17.5	+45	+8	-36	-0.2	-66	-58	+154
2	Standard Timing, 10% EGR, 150°F Aftercooling	1.02	0.20	2.98	0.414	0.021	2.8	20.9	+23	-23	-39	-1.9	-74	-39	+204
3	5° Retarded Timing, 10% EGR, No Aftercooling	1.58	0.29	2.78	0.443	0.058	4.5	10.0	+90	+11	-44	+5.0	-29	-2	+45
4	5° Retarded Timing, 10% EGR, 150°F Aftercooling	1.68	0.64	2.44	0.430	0.107	5.9	6.5	+102	+146	-50	+1.9	+30	+28	-6
	Baseline, Engine No. 28[d]	7.94	1.62	15.9	0.476	0.088	4.2	2.8	-	-	-	-	-	-	-
1	Experimental Injectors, Standard Timing	5.45	0.72	10.5	0.492	0.059	3.4	4.5	-31	-55	-34	+3.4	-33	-19	+61
2	Experimental Injectors, 3.4° Retarded Timing	9.78	0.88	7.91	0.491	0.125	4.3	18.2	+23	-45	-50	+3.2	+42	+5	+550

[a] Brake Specific Fuel Consumption

[b] Odor intensity in diesel intensity (DI) units determined with samples diluted 400:1 for engines 15 and 19, and 100:1 for all other engines

[c] Referenced to baseline values

[d] See Tables 3-2 through 3-7

TABLE 4-7 (continued)

Control System	Description of Engine Adjustment and Accessory Hardware	13-Mode Cycle						Lug-Down Smoke, % Opacity	Percentage Change[c]						
		Emissions, g/bhp-hr			BSFC,[a] lb/bhp-hr	Aldehydes, g/bhp-hr	Odor Intensity, DI Units[b]								
		CO	HC	NO_2					CO	HC	NO_x	BSFC[a]	Aldehydes	Odor	Smoke
	Baseline, Engine No. 6[d]	5.93	4.41	7.41	0.504	0.23	4.5	12.5	-	-	-	-	-	-	-
1	Standard Timing, 10% EGR (EGR cut off at full load)	5.71	3.22	6.41	0.498	0.17	4.8	11.7	-4	-27	-13	-1.2	-26	+7	-6
2	Standard Timing, 15% EGR (EGR cut off at full load)	6.01	2.85	6.00	0.512	0.23	5.1	11.7	+1	-35	-19	+1.6	0	+13	-6
3	2° Retarded Timing, 10% EGR (EGR cut off at full load)	5.68	4.12	5.95	0.510	0.33	4.7	10.1	-4	-7	-20	+1.2	+43	+5	-19
4	2° Retarded Timing, 15% EGR (EGR cut off at full load)	6.05	3.79	5.30	0.520	0.33	5.4	10.1	+2	-14	-28	+3.2	+43	+20	-19
5	Standard Timing, UOP Catalysts	0.68	0.53	8.47	0.514	0.046	2.6	14.0	-89	-88	+14	+2.0	-80	-42	+12
6	Standard Timing, 10% EGR, UOP Catalysts, (EGR cut off at full load)	0.68	0.55	6.80	0.520	0.065	2.3	13.4	-89	-88	-8	+3.2	-72	-49	+7
	Baseline, Engine No. 16[d]	4.48	2.62	15.1	0.490	0.14	5.9	7.6	-	-	-	-	-	-	-
1	3° Retarded Timing, Special Pump and Nozzle Kit, 200°F Aftercooling	3.70	1.87	10.4	0.502	0.12	5.1	7.7	-17	-28	-31	+2.4	-14	-13	+1
2	3° Retarded Timing, Special Pump and Nozzle Kit, 150°F Aftercooling	3.45	1.76	10.3	0.507	0.13	5.0	6.6	-23	-33	-32	+3.5	-7	-15	-13
3	3° Retarded Timing, Standard Pump and Nozzles, 150°F Aftercooling	3.76	2.60	8.82	0.495	0.18	7.4	7.4	-16	-1	-41	+1.0	+29	+25	-2

[a] Brake Specific Fuel Consumption

[b] Odor intensity in diesel intensity (DI) units determined with samples diluted 400:1 for engines 15 and 19, and 100:1 for all other engines

[c] Referenced to baseline values

[d] See Tables 3-2 through 3-7

reduction in NO_x, accompanied by only a two percent penalty in fuel consumption. A pair of Engelhard noble metal catalysts incorporated in the engine exhaust reduced CO by about 70 percent and HC by 38 to 58 percent. The observed variations in aldehydes, odor intensity, and peak smoke levels are quite moderate. Control System No. 2, which consisted of three degrees timing retard, five percent EGR, and Engelhard catalysts, appeared to be the most cost-effective configuration for this particular engine. It reduced NO_x by 40 percent at the expense of only a 1.8 percent loss in fuel consumption.

Test data from Engine No. 24 indicate that the Emission Control Systems No. 2 and No. 4 are most effective from a NO_x emission reduction and SFC point of view. In the case of System No. 2, NO_x was reduced by about 40 percent, with a concomitant loss in SFC of about two percent. System No. 4 resulted in a 50-percent reduction in NO_x, but the specific fuel consumption increased by about two percent. It is estimated that a 45-percent reduction in NO_x could be achieved without any loss in fuel economy by means of an emission control system consisting of ten percent EGR, intake air cooling to 150°F, and three degrees timing retard (instead of the five degrees applied in System No. 4). In this case, aldehyde and odor emissions would not change very much from the levels of the unmodified engine, but smoke would increase, perhaps by as much as 100 percent.

Incorporation of emission control System No. 2 into Engine No. 28 (3.4-degrees timing retard and experimental injectors) resulted in a 50-percent improvement in NO_x, accompanied by a 3.2-percent loss in fuel economy, and substantial increases in aldehyde and smoke emissions. However, it should be noted that the lugdown smoke of the baseline engine was lower than for the other engines tested in this program.

The NO_x reductions achieved with the various emission control systems on Engine No. 6 are lower than for the other engines.

The largest reduction (28 percent) was realized with Emission Control System No. 4, utilizing two degrees timing retard and 15 percent EGR, at the expense of a 3.2-percent increase in specific fuel consumption. The use of a pair of UOP noble metal catalysts caused reductions in CO and HC of about 88 percent. However, it should be noted that these were fresh catalysts and durability testing would be required to verify the performance of these catalysts over a long time period. With catalysts, the aldehyde and odor emissions were reduced substantially, relative to the baseline levels, while the smoke intensity remained essentially unchanged.

Engine No. 16 was tested with three different emission control systems incorporating a special pump and nozzle kit, aftercooling, and injection timing modifications. The special pump was equipped with a built-in puff limiter for acceleration smoke control and an experimental idle feature, and the special nozzles had a wider spray angle than the standard nozzles. Minimum NO_x emissions were obtained with Control System No. 3, utilizing three degrees timing retard, intake air cooling to $150^{\circ}F$, and the standard pump and nozzles. In this case, NO_x was reduced by 41 percent, while specific fuel consumption increased by only about one percent. Aldehydes, odor, and smoke were not affected much by the different emission control systems.

Emission versus fuel consumption correlations projected by Cummins for future diesel engines are shown in Figure 4-30 (Ref. 4-26). As indicated, the emissions of the current production engines might be reduced to the 5 g/bhp-hr (HC + NO_x) level specified by California for 1977 model year trucks by incorporating three sets of engine modifications, designated Phase I, Phase II, and Phase III. Phase I consists of smaller production tolerances, retarded fuel injection timing, higher engine compression ratio, a new turbocharger, and intercooling in the lower BMEP range. Phase II includes the Phase I modifications plus a larger camshaft and variable injection timing.

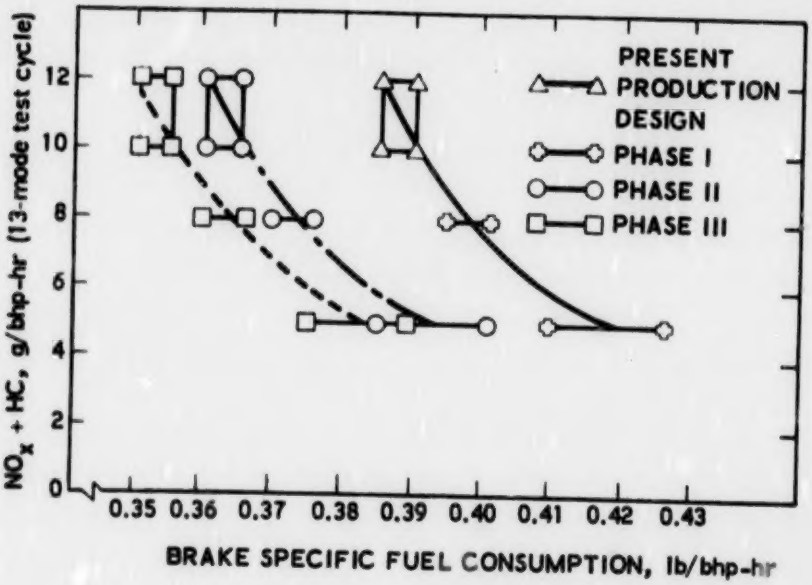


Figure 4-30. Projected effect of emission control systems on emissions and specific fuel consumption (Ref. 4-26)

Finally, in Phase III, a new injector will be utilized in conjunction with the Phase II modifications.

Referring to Figure 4-30, the fuel economy of the present production engines can be achieved at the 5 g/bhp-hr emission level by incorporating the Phase II or Phase III modifications. Conversely, at the 10 to 12 g/bhp-hr emission level typical of many current open-chamber diesels, the fuel consumption of the Phase III engine can be reduced by about ten percent. In these systems, reduction in NO_x is accomplished by the timing retard and improved aftercooling, while the lower HC levels are primarily due to the higher engine compression ratio and variable timing. The design of these engine modifications has not yet been finalized. Based on some problems experienced during testing of the Phase I hardware, Cummins is very concerned about rapidly meeting acceptable durability and reliability standards for these new engines. However, with normal development time, Cummins is confident that these problems can be resolved.

Similar design modifications are being considered by other diesel engine manufacturers to meet future emission standards. For example, General Motors intends to meet the 1975 California standards for heavy-duty vehicles by incorporating a new turbocharger, an air-to-water aftercooler, and retarded injection timing. No information is available relative to the fuel consumption of these engines.

4.1.4.2 Steady-state Tests

The combined effects of injection timing and air-swirl modifications were evaluated by Khan et al. (Ref. 4-10), using a single-cylinder, open-chamber, naturally aspirated diesel. Increasing the air swirl makes the smoke versus timing curve less sensitive to timing changes and retards the timing for optimum efficiency. In addition, by using the optimum combination of air swirl and timing, NO_x can be reduced without sacrificing fuel economy. For example, changing the swirl ratio from 2 to 8 and the injection timing from 17°BTDC to about 6°BTDC resulted in a reduction in NO_x from 1350 ppm to 1000 ppm, an improvement of about 25 percent, with no loss in fuel consumption.

Emission and specific fuel consumption factors of a turbocharged, divided-chamber diesel engine tested by the Bureau of Mines (Ref. 4-3), without and with incorporation of an emission control system, are presented in Figure 4-31. The particular emission control system used on this engine consisted of ten percent EGR, combined with five-degree injection timing retard. At rated speed and full load, NO_x was reduced by about 50 percent, at the expense of a six percent loss in specific fuel consumption. At the intermediate speed of 1600 rpm, the NO_x reduction at full load was about 60 percent, while the loss in specific fuel consumption was over ten percent. The NO_x effectiveness of

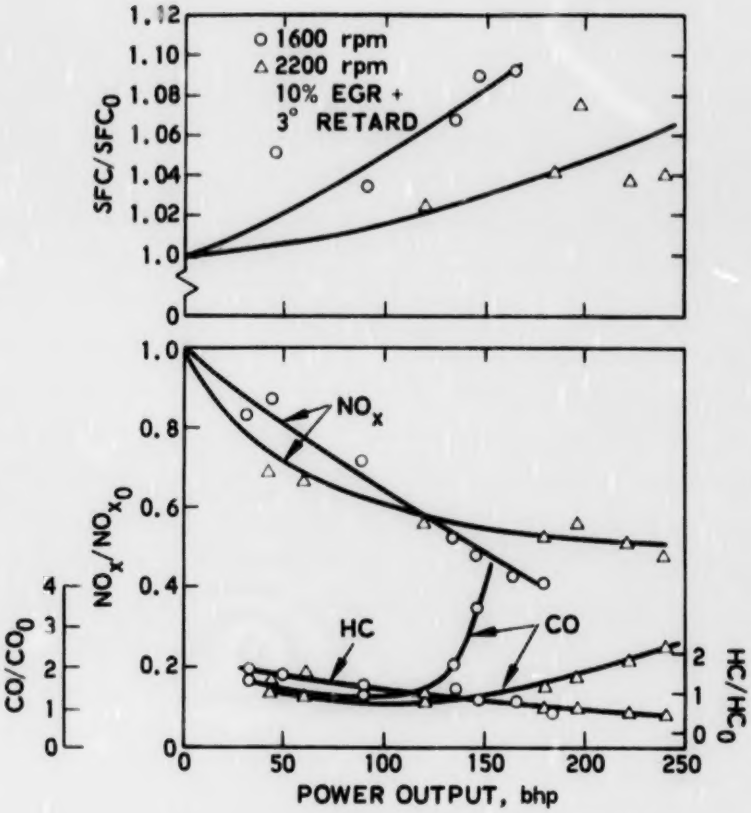


Figure 4-31. Effect of 10 percent EGR and 5° injection timing retard on specific fuel consumption and emissions (Engine No. 24)

the emission control system decreased with decreasing load, but the loss in specific fuel consumption decreased also, for both operating speeds. The effect of the control system on HC was essentially independent of speed and varied by about ±50 percent over the load range evaluated in this program. With emission control, CO is higher, particularly in the high load regime.

Data published by Cooper-Bessemer for its KSV-12 diesel engine operated on both No. 2 diesel fuel and on natural gas and fitted with several different combinations of emission control devices/ techniques are listed in Table 4-8 (Ref. 4-6). The control systems evaluated by Cooper-Bessemer include various combinations of timing retard, increased air flow, lower intake air temperature, and water injection into the intake manifold. In the case of diesel fuel, lowering the manifold temperature from 130°F to 100°F, combined with four degrees timing retard, resulted in a 21-percent reduction in NO_x and a slight improvement in specific fuel consumption. Increasing the air flow rate had no effect on NO_x, but provided further improvement in the specific fuel consumption of the engine. Adding water injection caused an additional six percent reduction in NO_x at the expense of an increase in specific fuel consumption back to the level of the uncontrolled engine. When operating with natural gas, these emission control measures were more effective in terms of reducing NO_x, but the specific fuel consumption increased by as much as 3.6 percent. In this case, the observed variations in HC and CO were not very large. As expected, CO increased at the lower air intake temperature, but decreased again as more air was added.

4.2 SPARK IGNITION ENGINES

4.2.1 Modification of Engine Operating Conditions

A number of important engine operating parameters have been identified which have a strong effect on the NO_x, HC, and CO emissions from spark ignition engines. These include air-fuel ratio; ignition timing; compression ratio; mixture and coolant temperatures; engine speed and load; valve timing, exhaust backpressure; and combustion chamber deposits. These parameters are discussed in the following subsections.

4.2.1.1 Air-Fuel Ratio

As shown in Figure 4-32, the air-fuel ratio of the combustible mixture has a very pronounced effect on the NO_x, HC, and

TABLE 4-8. EFFECT OF COMBINED EMISSION CONTROL TECHNIQUES ON DIESEL ENGINE EMISSION AND SPECIFIC FUEL CONSUMPTION (Ref. 4-6)

Fuel	Injection Timing	Air Intake Tempera-ture, °F	Air Flow Rate	Water Injection, gpm	Mass Emissions g/bhp-hr			BSFC,[a] Btu/bhp-hr	Emission Ratios			
					HC	CO	NO_x		$\frac{HC}{HC_0}$[b]	$\frac{CO}{CO_0}$	$\frac{NO_x}{NO_{x_0}}$	$\frac{BSFC^a}{BSFC_0}$
No. 2 Diesel	Standard	130	Standard	0	0.13	3.85	10.99	6677	1.0	1.0	1.0	1.0
No. 2 Diesel	-4°	100	Standard	0	0.20	4.09	8.71	6636	1.54	1.06	0.79	0.994
No. 2 Diesel	-4°	100	+6%	0	0.20	3.39	8.66	6583	1.54	0.88	0.79	0.986
No. 2 Diesel	-4°	100	+6%	0.5	0.20	3.67	7.98	6664	1.54	0.95	0.73	0.998
Natural Gas	Standard	130	Standard	0	5.16	4.50	8.96	6340	1.0	1.0	1.0	1.0
Natural Gas	-4°	100	Standard	0	3.20	6.45	6.63	6440	0.62	1.43	0.74	1.016
Natural Gas	-4°	100	+10%	0	7.39	3.44	5.29	6530	1.43	0.76	0.59	1.030
Natural Gas	-4°	100	+10%	0.3	7.76	3.17	5.27	6568	1.50	0.70	0.59	1.036

[a] Brake Specific Fuel Consumption

[b] Subscript zero refers to baseline values.

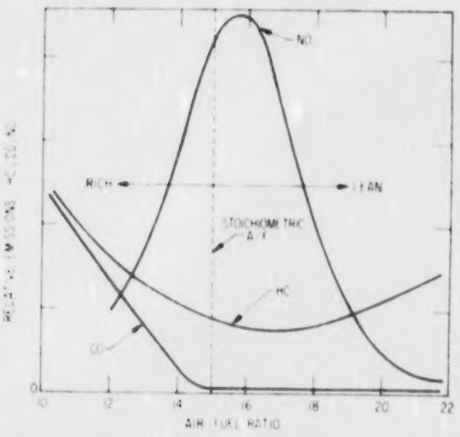


Figure 4-32. Effect of air-fuel ratio on emission levels, gasoline spark-ignition engine

CO exhaust concentrations emitted from spark ignition engines. Adjustment of the air-fuel ratio to a fuel rich mixture results in low NO_x emission, at the expense of an increase in CO and HC, as well as fuel consumption. This approach is usually taken in combination with corrective emission control measures such as thermal reactors and/or catalytic converters.

In conventional gasoline engines, adjustment of the air-fuel ratio to a fuel lean mixture is limited to slightly leaner than stoichiometric (approximately 15.5 to 16). At this setting, the HC and CO concentrations approach a minimum whereas, the concentration of NO_x reaches a maximum. Substantial further leaning of the air-fuel mixture, which would result in the lowest concentration of all three contaminants, appears feasible only by means of mixture stratification or improved carburation and fuel injection techniques.

It should be noted that the smog-forming potential of the exhaust gases depends not only on the quantity of HC and NO_x emitted but also on the reactivity of the various HC compounds (Refs. 4-27, 4-28, and 4-29). Among the many HC types emitted, the olefines have the highest reactivity while the paraffines have the lowest reactivity. The effect of air-fuel ratio on reactivity is illustrated in Figure 4-33 (from Ref. 4-28) showing chromatograph data from a single-cylinder spark-ignition gasoline engine. As indicated, the total hydrocarbon concentration decreased rapidly with leaning of the mixture and reached a minimum at air-fuel ratio of about 16. Conversely, the total HC reactivity index (smog-forming potential) showed much less variation over the range of air-fuel ratios investigated. The apparent benefit of leaning the air-fuel mixture (up to A/F = 17) for reduction of unburned hydrocarbons is considerably diminished by the relatively modest reduction of the total reactivity index of the exhaust. In addition to air-fuel ratio the total reactivity index of the exhaust depends on the engine design and the type of fuel used. The effect of air-fuel ratio on the fuel economy of an automobile equipped with a V-8 spark ignition engine and cruising at 30 mph is shown in Figure 4-34 (Ref. 4-30). In this case the best fuel economy is achieved at an air-fuel ratio of about 16, which coincides with minimum HC.

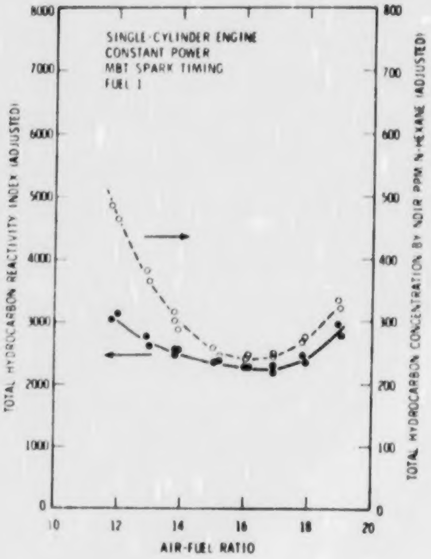


Figure 4-33. Effect of air-fuel ratio on reactivity index and concentration measured by infrared analyzer (Ref. 4-28)

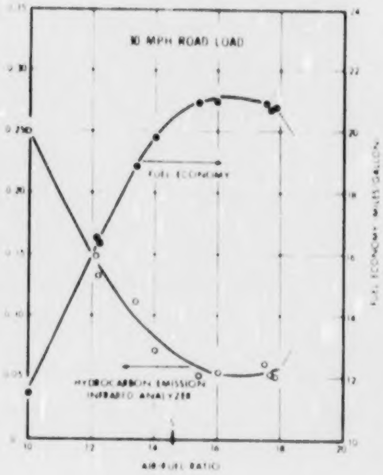


Figure 4-34. **Effect of air-fuel ratio on exhaust hydrocarbon emission and fuel economy in car at 30 mph roadload (Ref. 4-30)**

The effect of air-fuel ratio on the emissions of a Cooper Bessemer GMVA-8, two-stroke stationary spark ignition gas engine, rated at 1080 bhp, 300 rpm and 82.5 bmep is illustrated in Figure 4-35 (Ref. 4-31). In these tests the air-fuel ratio was varied over a narrow range (A/F = 22 to 26) by changing the scavenging air flow rate. Again, NO_x decreased with increasing air-fuel ratio while CO remained unchanged. However, unlike automotive engines, HC showed no variation over the range of air-fuel ratios evaluated by Cooper Bessemer. The specific fuel consumption has a minimum at the baseline operating conditions indicated in the figure, while the firing pressure and the spark plug gasket and cylinder exhaust temperatures decline steadily with increasing air-fuel ratio.

4.2.1.2 Ignition Timing

Ignition timing has a strong effect on the HC and NO_x emissions of spark ignition engines. In general these species decrease substantially with increasing ignition timing retard.

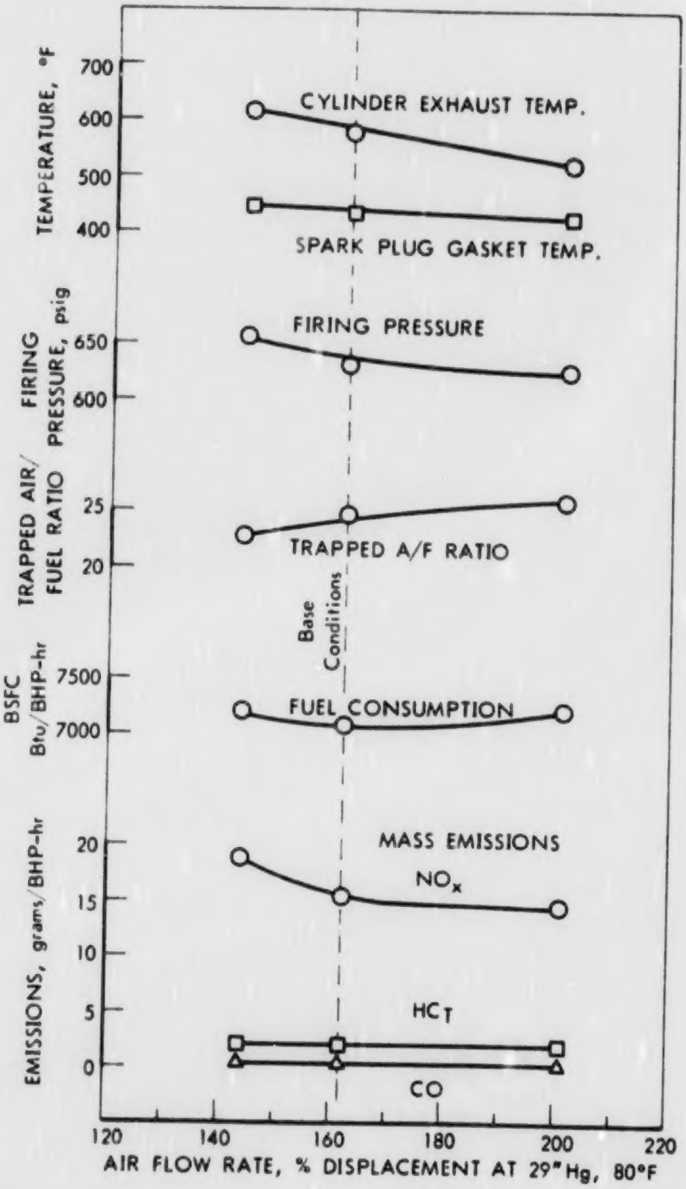


Figure 4-35. Effect of air flow rate on engine emissions and performance - Cooper Bessemer GMVA-8 2-stroke atmospheric spark-gas engine; 1080 bhp at 300 rpm, 82.5 bmep, base conditions - (Ref. 4-31)

From several experimental and theoretical studies (Refs. 4-32 and 4-33) it can be concluded that the nitric oxide concentration in the exhaust correlates reasonably well with the peak combustion temperature in the engine. This is illustrated in Figure 4-36 (Ref. 4-32) showing a plot of measured peak cycle temperature and NO_x versus spark advance (air-fuel ratio 11:1). The general trends shown in the figure support the view that the reduction of nitric oxide by means of spark retard is due to the attendant reduction of the peak combustion temperature. The combined effect of spark retard and air-fuel ratio on the NO_x concentration in the exhaust of a CRF engine is shown in Figure 4-37 (Ref. 4-34). As indicated, the effect of ignition timing is very pronounced for lean mixtures (air-fuel ratio >15) and rather modest for rich mixtures (air-fuel ratio <14). Similar results were reported by Huls, et al (Ref. 4-35).

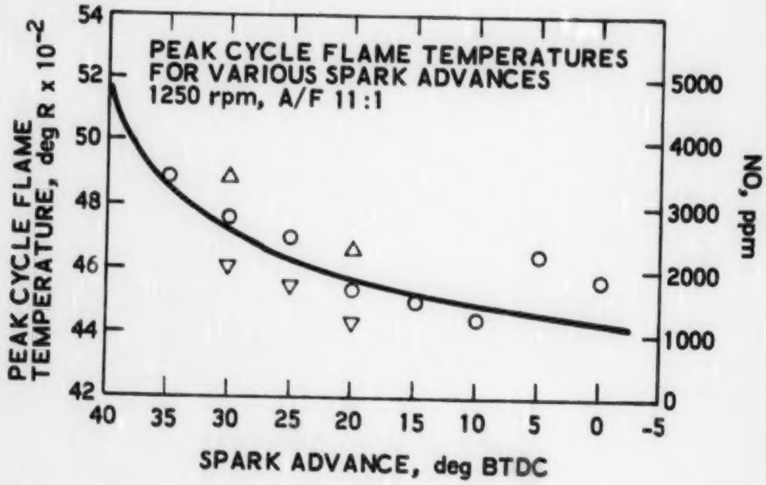


Figure 4-36. Correlation between peak cycle temperature and NO concentration (Ref. 4-32)

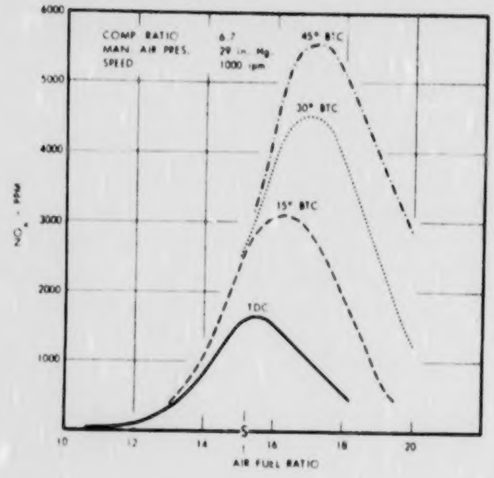


Figure 4-37. Effect of air-fuel ratio and of spark timing on oxides of nitrogen (Ref. 4-34)

The effect of spark-ignition timing on the HC emissions is more complex because the reactivity index varies significantly with air-fuel ratio. Figure 4-38 shows the effect of ignition timing on the total hydrocarbon emission for rich mixtures, as determined by chromatography (Ref. 4-28). The HC emission obtained for that engine at air-fuel ratio of 16 is presented in Figure 4-39. In this case, the acetylene, total olefin, and total hydrocarbon concentrations increased as the spark timing was retarded, while the total paraffin and total aromatic concentrations decreased. From the standpoint of reducing smog-forming potential, these are not the desired results.

The effect of spark ignition timing on exhaust hydrocarbon emission and the attendant fuel economy for a test car at 30 mph (air-fuel ratio = 16) has been evaluated by Jackson et al (Ref. 4-30). Based on the available test data it is concluded that retarding the ignition timing has an adverse effect on fuel economy and therefore may be practical only if applied to a limited extent during certain modes of engine operation.

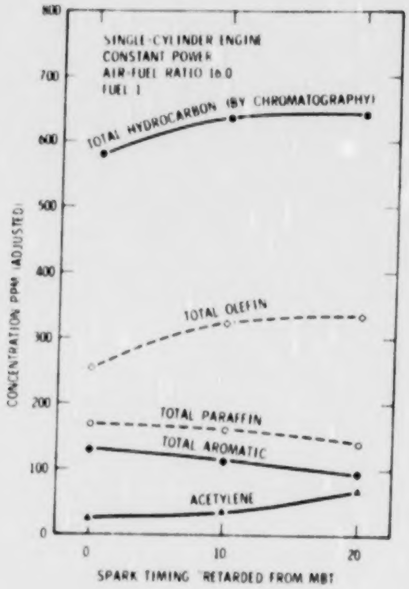


Figure 4-38. Effect of spark timing on hydrocarbon composition by class at rich air-fuel ratio (Ref. 4-28)

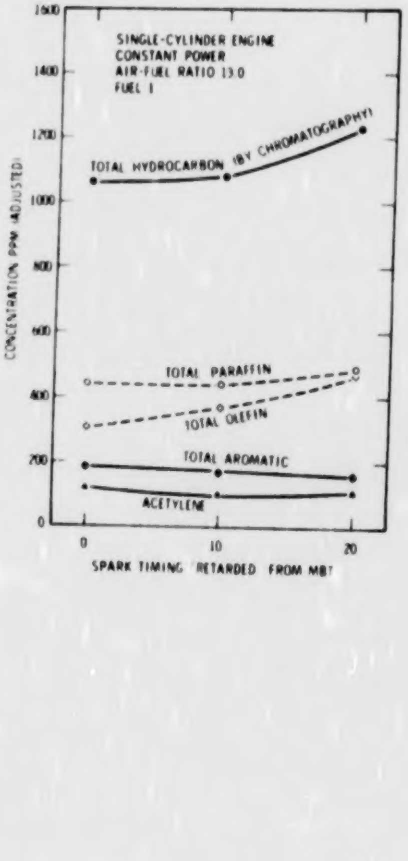


Figure 4-39. Effect of spark-timing on hydrocarbon composition by class at lean air-fuel ratio (Ref. 4-28)

Figure 4-40 presents the effect of ignition timing on the exhaust emissions of a Cooper Bessemer GMVA-8, two-stroke stationary engine using gaseous fuel (Ref. 4-31). Retarding the ignition timing from the standard setting of ten degrees BTDC to four degrees, results in a 15 percent reduction in NO_x, accompanied by very small variations in HC and CO. However, the specific fuel consumption of the engine increases substantially, from about 7070 Btu/bhp-hr to about 7500 Btu/bhp-hr. At the same time, the firing pressure decreases, whereas the cylinder exhaust gas temperature increases.

4.2.1.3 Mixture Temperature

The temperature of the air-fuel mixture supplied to the engine has a noticeable effect on the concentration of the contaminants in the exhaust. Several factors determine the temperature of the mixture prior to its admission into the engine cylinder including the atmospheric conditions of the intake air (temperature, density and humidity), the heat capacity and heat of vaporization of the fuel, and the amount of heat transferred to the mixture in the engine intake manifold.

The effect of atmospheric conditions is relatively small, but important enough to require numerical correction factors to the measured emission data of nitric oxide, in order to make possible an unbiased comparison of test data, taken under varying conditions of atmospheric humidity (Ref. 4-33).

The heat transfer and the fuel evaporation process in the engine inlet manifold affects the mixture temperature more substantially. Figure 4-41 (Ref. 4-36) presents the effect of inlet manifold heating on mixture temperature and on the resulting emissions of NO_x, HC, and CO. The tests were performed on a car equipped with a V-8 spark ignition engine. The emissions were measured at car cruising speeds up to 70 mph, with the inlet manifold heated in a conventional way by exhaust gases passing through the manifold cross-over

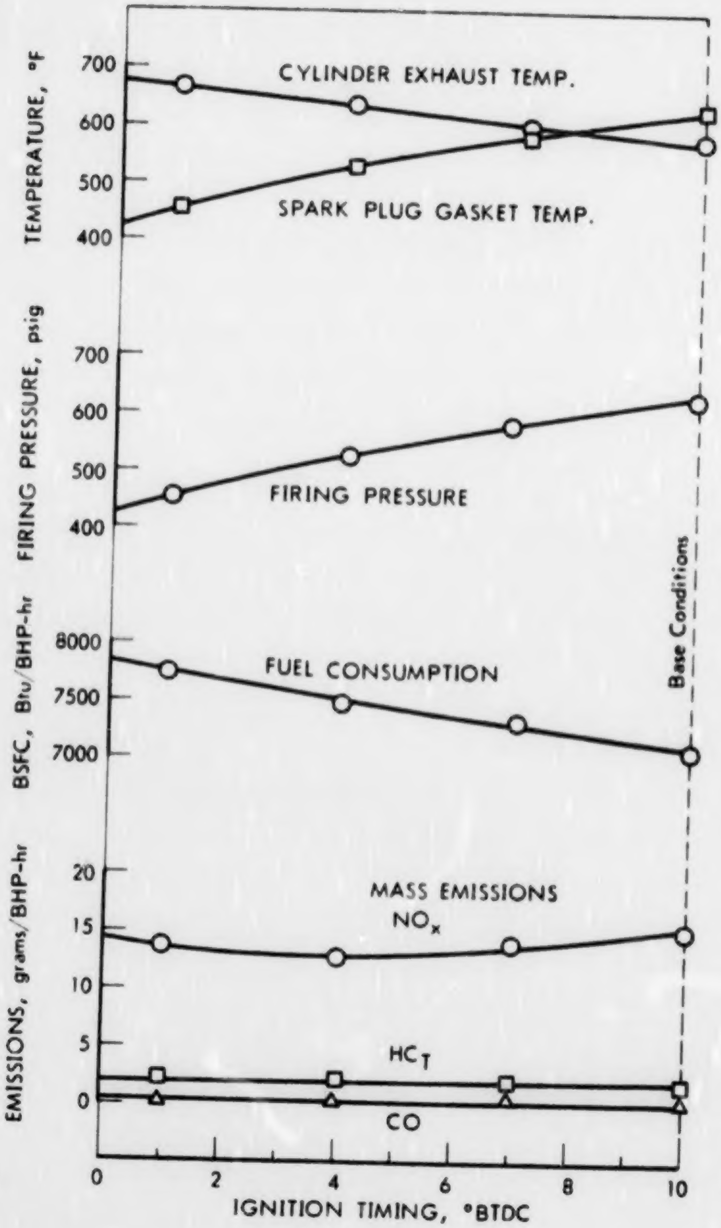


Figure 4-40. Effect of ignition timing on engine emissions and performance - Cooper Bessemer GMVA-8 2-stroke atmospheric spark-gas engine, 1080 bhp at 300 rpm, 82.5 bmep, base conditions (Ref. 4-31)

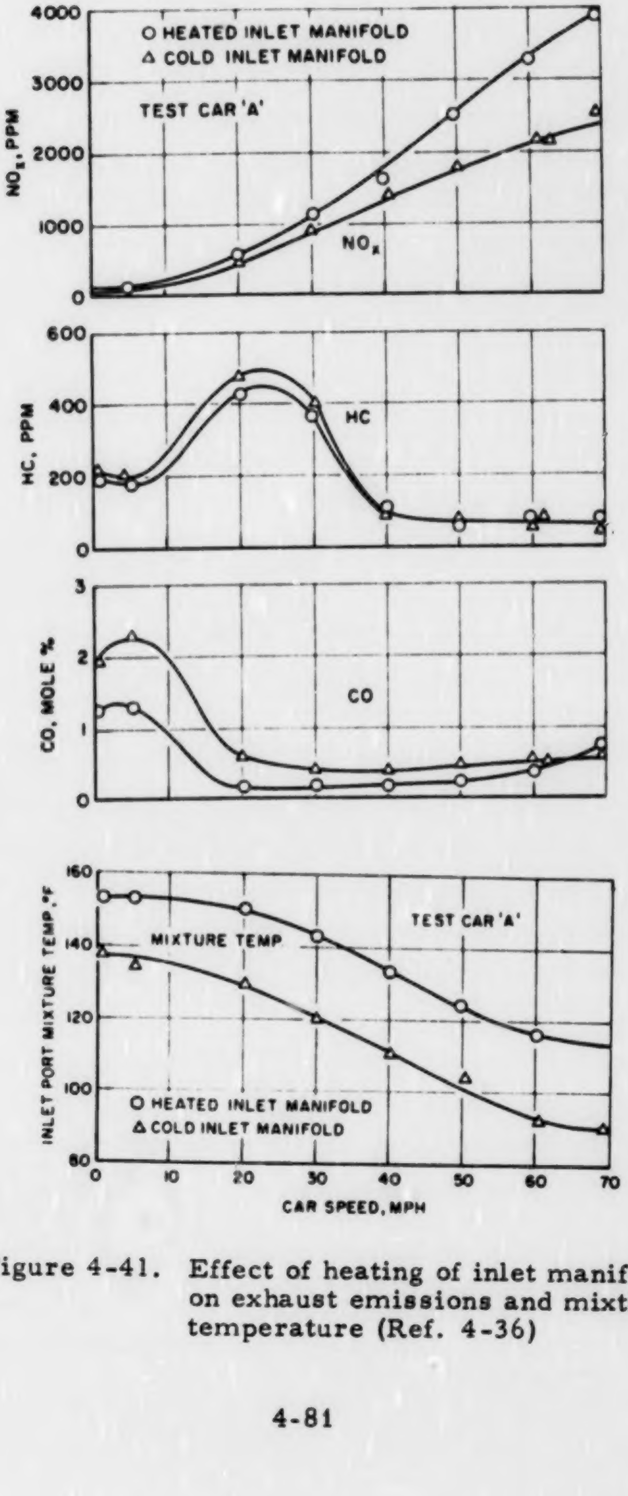


Figure 4-41. Effect of heating of inlet manifold on exhaust emissions and mixture temperature (Ref. 4-36)

passage. For the tests with cold inlet manifold, the cross-over passage was blocked off on both sides of the manifold. As indicated in Figure 4-41 a mixture temperature drop of approximately 20°F (with blocked-off heating passage) resulted in a substantial reduction (20 to 40 percent) of NO_x. However, hydrocarbons increased slightly (at low speeds only) and carbon monoxide increased substantially throughout the test range. The observed reduction in NO_x is attributed to the lower peak combustion temperature which results from the lower intake temperature.

On the other hand the higher HC and CO emissions are considered to be due to fuel maldistribution and wall quench effects (Ref. 4-37). The magnitude of these effects depends to some degree on the engine inlet manifold and fuel system design and the type of fuel used. In general, cooling of the mixture decreases the tendency to engine knock (Ref. 4-38). No information is currently available on the effects of mixture temperature on the reactivity index of the exhaust gas.

Figure 4-42 (Ref. 4-31) shows the effect of inlet air temperature on the exhaust emissions of a Cooper Bessemer GMVA-8, two-stroke stationary gas engine. In general terms, the observed trends are quite similar to those of automotive spark ignition engines, but the magnitude of the changes varies considerably. For example, a reduction of the manifold temperature from the standard setting of 130 to 100°F caused a decrease in NO_x from about 15 g/bhp-hr to 11 g/bhp-hr, a reduction of about 27 percent. This was accompanied by a slight increase in air-fuel ratio and specific fuel consumption. The HC and CO emissions remained essentially unchanged over the range of temperatures investigated by Cooper Bessemer.

4.2.1.4 Coolant Temperature

For satisfactory engine operation the temperature of the engine must be maintained within predetermined limits. If the

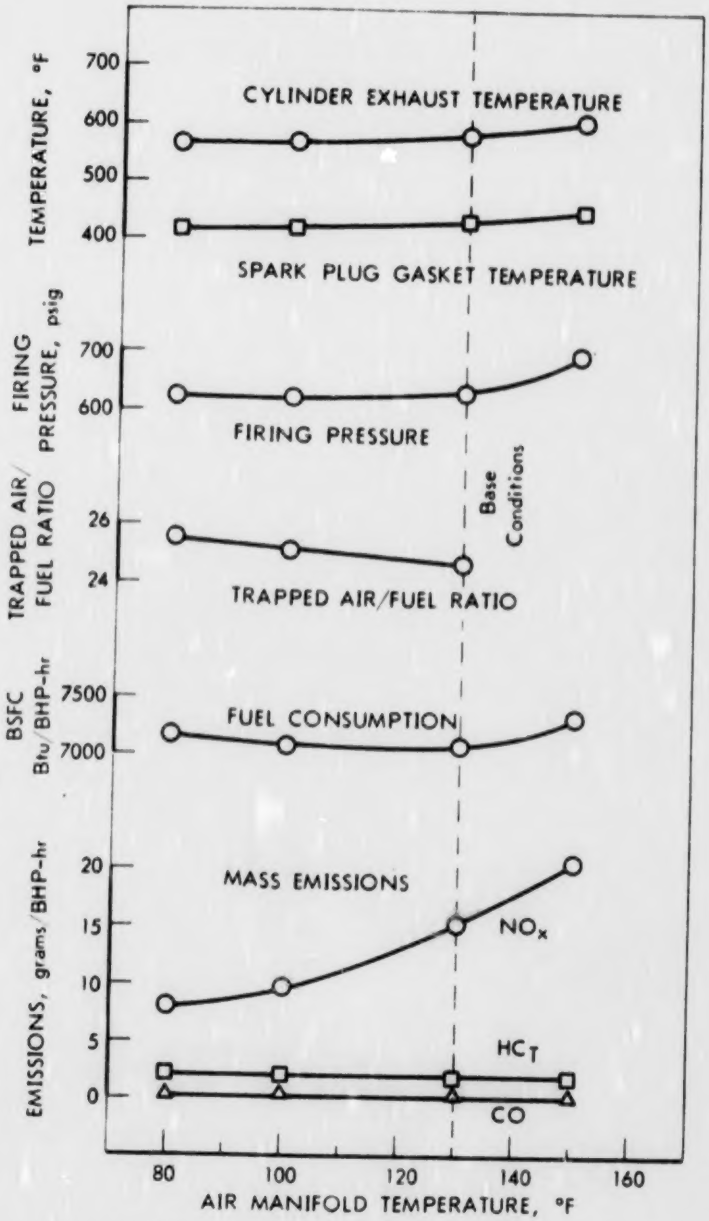


Figure 4-42. Effect of air manifold temperature on emissions and performance - Cooper Bessemer GMVA-8 two-stroke atmospheric spark-gas engine; 1080 bhp at 330 rpm, 82.5 bmep, base conditions (Ref. 4-31)

engine becomes excessively hot then preignition and "engine knock" may develop and adversely affect its performance. Varying the engine temperature within the permissible limits changes the degree of preheating of the combustible mixture prior to ignition, hence affecting the exhaust emissions, as previously discussed in Section 4.2.1.3. Furthermore, the combustion chamber wall temperature affects the thickness of the quench layer adjacent to the chamber walls. A portion of this quench layer which contains unburned fuel is exhausted during the exhaust stroke (Ref. 4-39).

The relationship between the combustion chamber surface temperature and the HC emissions has been the subject of intensive studies. Figure 4-43 presents the effect of the average combustion chamber surface temperature on the concentration of exhaust hydrocarbons (Ref. 4-40). The data shown are for a production 6-cylinder, 230 cu in. spark ignition engine having a compression ratio of 8.5. The engine was operated on isooctane fuel using a constant air-fuel ratio of 14, constant inlet air temperature (123°F) and constant exhaust back pressure. As indicated, HC decreased markedly with increasing surface temperature. Similar trends were observed at other engine speed and load settings.

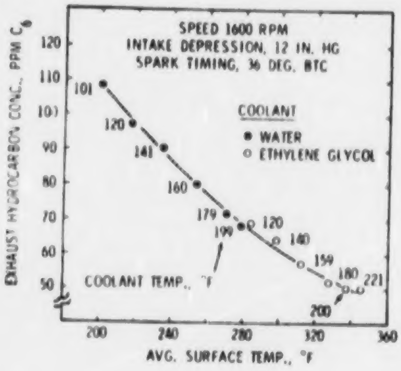


Figure 4-43. Effect of average combustion chamber surface temperature on hydrocarbon emission (Ref. 4-40)

Test data published by Daniel (Ref. 4-41) for a single-cylinder engine indicate that the effect of coolant temperature variations on the concentrations of the various HC species is very similar. As a result the exhaust reactivity index is believed to be proportional to the total HC concentration in the exhaust.

The effects of coolant temperature on the NO_x, HC, and CO emissions, as determined on a single cylinder gasoline engine, are shown in Figure 4-44 (Ref. 4-42). In addition, the figure shows the effects of spark timing and air-fuel ratio. As expected, the specific mass emission of NO_x (g/ihp-hr) increased noticeably with increasing coolant temperature. At the stoichiometric fuel-air ratio, changing the coolant temperature from 370 to 212°F resulted in a 20-percent reduction in NO_x, accompanied by a 30-percent increase in HC and some reductions in CO. In these tests the indicated mean effective pressure increased somewhat with decreasing coolant temperature, while the specific fuel consumption showed a tendency to decrease, particularly for retarded spark settings. Unfortunately this investigation was limited mostly to tests with richer than stoichiometric air-fuel ratios and on the basis of the data published it cannot be ascertained whether similar trends would apply also to lean air-fuel ratios.

4.2.1.5 Engine Speed

Relatively little information has been published on the effects of engine speed on exhaust emissions. Figure 4-45 presents the concentration of nitric oxide as a function of engine speed (Ref. 4-34). The tests were carried out on a single-cylinder experimental engine and the mixture temperature, manifold pressure, ignition timing and compression ratio were kept constant. With rich mixtures, NO_x increased with increasing speed. Conversely, in the lean regime NO_x decreased sharply with increasing speed.

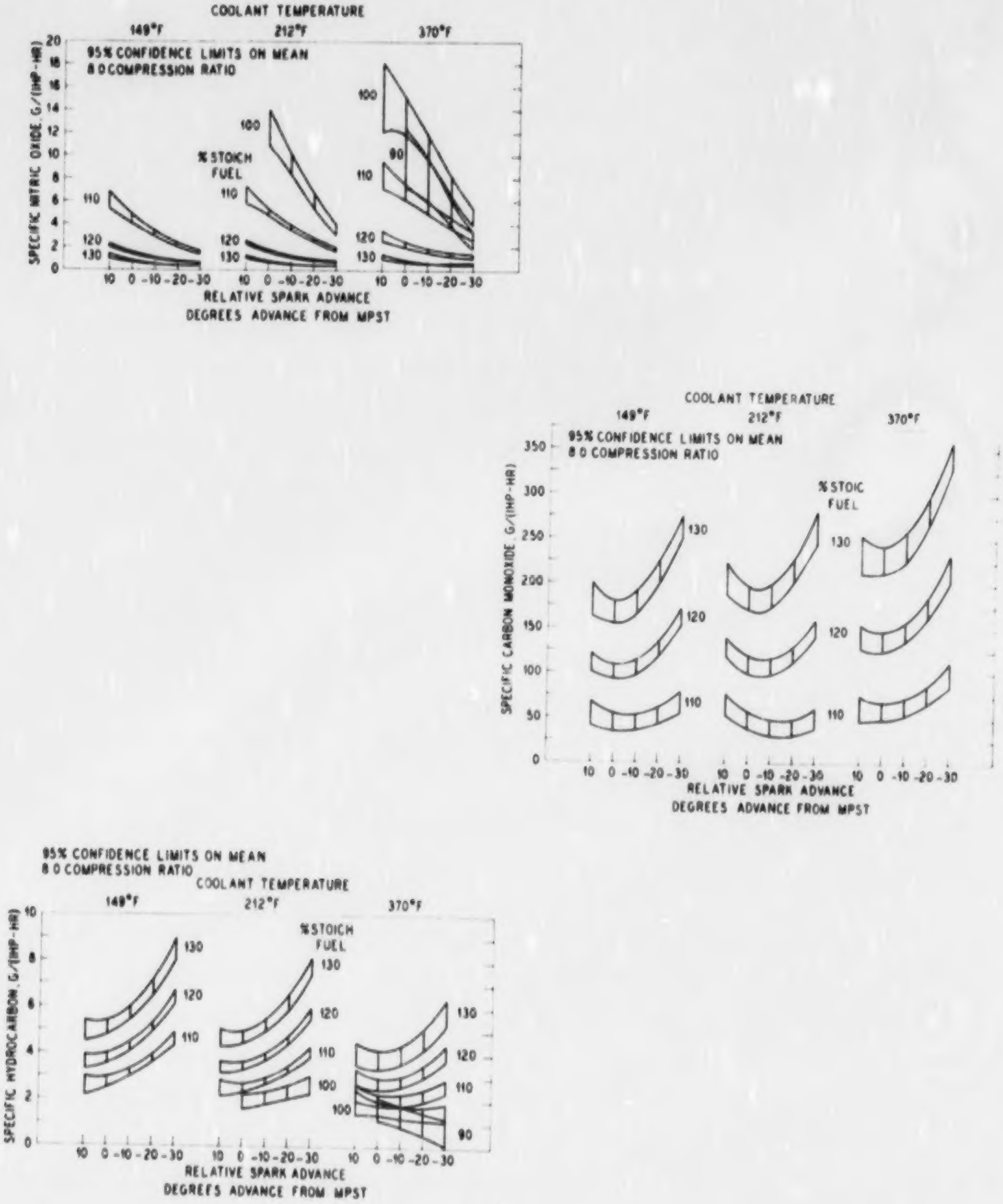


Figure 4-44. Effect of coolant temperature and spark advance upon indicated specific NO, HC, and CO emissions (Ref. 4-42)

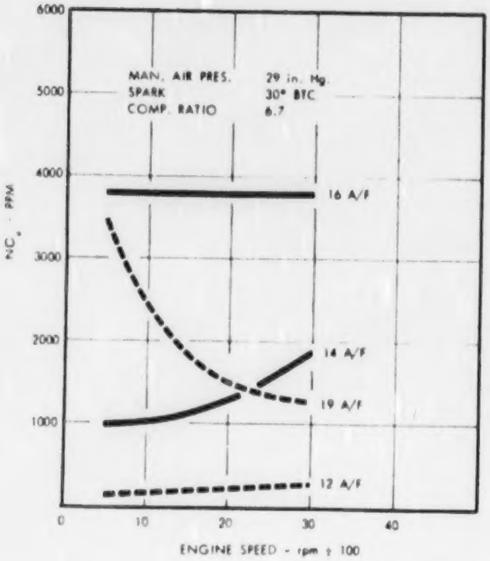


Figure 4-45. Effect of engine speed on NO_x concentration (Ref. 4-34)

In a more recent study the effects of engine speed and load on nitric oxide emissions were investigated with a single-cylinder engine operated on a rich indolene 30/air mixture (83 percent of stoichiometric) and with optimized ignition timing (Ref. 4-43). Again NO_x increased with increasing engine speed. This increase is attributed by the author to the variation in engine speed and load which caused a change in the degree of charge dilution (by residual gas). Furthermore, the author concluded that the effect of speed on NO_x was greatest at low engine load and the effect of load was greatest at low engine speed. Also, the speed and load effects appeared to be more pronounced in the case of larger valve overlap.

The effect of engine speed on HC concentration is shown in Figure 4-46 for three automotive spark ignition engines (144 CID, 292 CID, and 352 CID) operated at three different air-fuel ratios (Ref. 4-44). In all cases the concentration of exhaust hydrocarbons decreased with increasing rpm. This trend is even more apparent

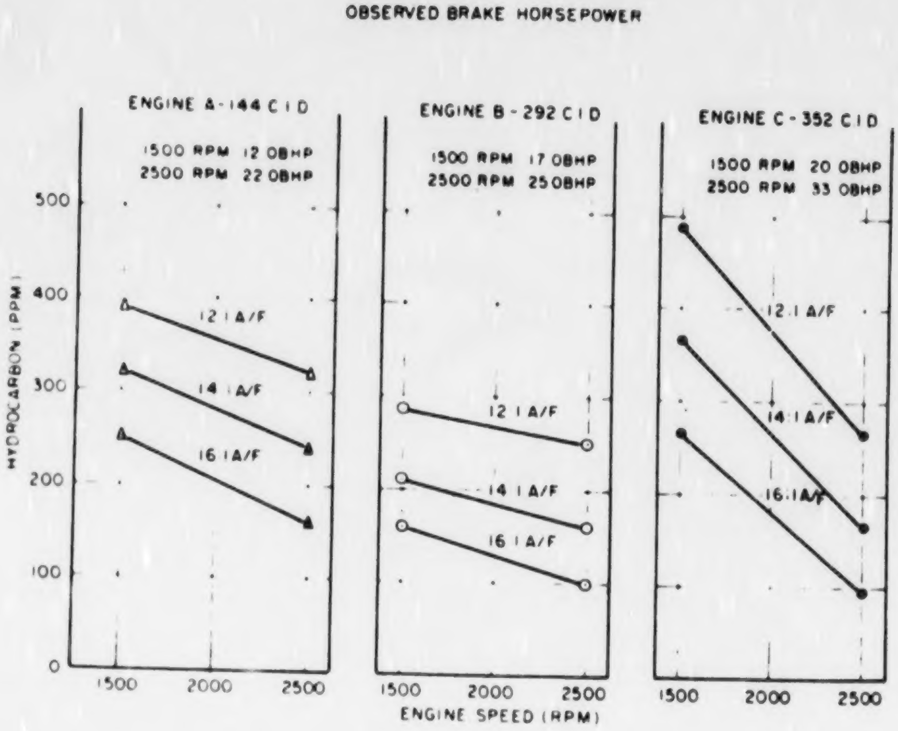


Figure 4-46. Effect of engine speed on HC concentration (Ref. 4-44)

when the emissions are expressed in terms of g/bhp-hr. Similar results were obtained by Daniel (Ref. 4-41).

While the decrease of exhaust hydrocarbons concentration with increasing engine speed is well known, it has been one of the most difficult phenomena to explain. In a study of the surface phenomena associated with exhaust hydrocarbon emissions (Ref. 4-45), a hypothesis is presented which relates the HC emissions to the piston top land clearance volume. The observed decrease in HC with increasing engine speed is attributed to an increase in the average cylinder surface temperature resulting in thermal expansion of the piston crown, hence a reduction of the top-land volume and the amount of ejected unburned hydrocarbons.

No effect of engine speed on exhaust carbon monoxide concentration has been reported.

Test results from a larger two-stroke stationary gas engine are presented in Figure 4-47 (Ref. 4-31). As indicated NO_x follows a trend similar to automotive spark-ignition engines. Rough engine operation was noted at 400 rpm, which might explain the sharp increase in HC. Increasing the engine speed from 300 rpm (standard speed) to 330 rpm resulted in a reduction in NO_x from about 15 g/bhp-hr to about 7 g/bhp-hr. This effect which is accompanied by a small increase in specific fuel consumption, is the direct result of a reduction in the brake mean effective pressure of the engine and a substantial increase in air-fuel ratio.

4.2.1.6 Valve Timing

Traditionally, the major objective of cam design, for the timed actuation of the engine valve system, is to provide the highest volumetric efficiency and the least charge dilution (by residual gases in the engine cylinder) over the entire operating range of the engine. Fulfillment of this objective results in good engine performance and fuel economy.

However, a change in valve timing from the optimal setting has a significant effect on exhaust emissions, particularly NO_x. Generally any change in timing of the inlet or exhaust valve, will result in an increase of the charge dilution by a larger amount of residual gases trapped in the engine cylinder. As a result the combustion temperature and the formation of NO_x are reduced.

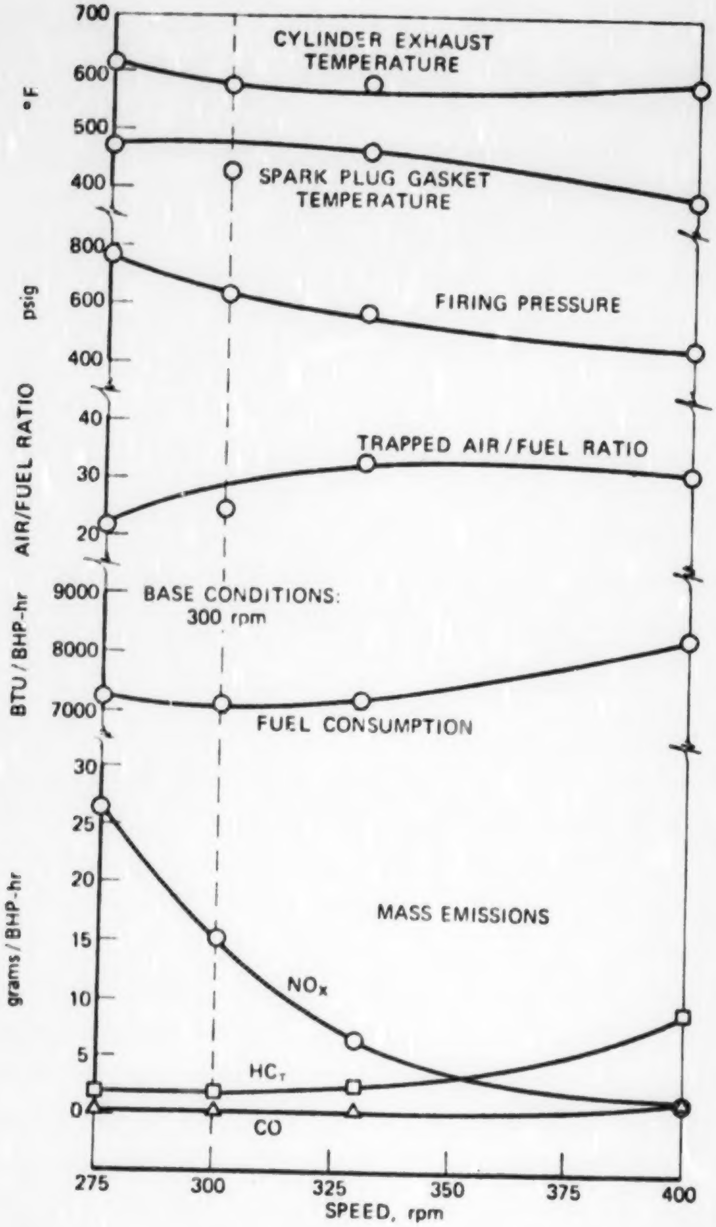


Figure 4-47. Effect of speed on emissions and performance - Cooper Bessemer GMVA-8 two-stroke atmospheric spark-gas engine, power output 1080 bhp, base conditions (Ref. 4-31)

The magnitude of the reduction depends also on the temperature of the internally recycled exhaust gas. In the case of early inlet valve opening the exhaust gases are partially discharged into the inlet manifold (intake reversed flow) where they are effectively cooled in contact with the relatively cold walls of the inlet manifold. During the piston intake stroke, the air-fuel mixture flow brings the cooled exhaust gases back into the engine cylinder, thus increasing the charge dilution (Ref. 4-46).

Since early inlet valve opening causes the exhaust system to be in communication with the low pressure intake system for a longer period of time, exhaust gases are returned to the cylinder through the exhaust valve.

The exhaust gas backflow originates from the end of the exhaust stroke and thus is expected to have larger hydrocarbon concentrations (Ref. 4-39). Retention of these hydrocarbon-rich quench gases is believed to be the main reason for the observed reduction in hydrocarbon emission, as indicated in Figure 4-48. The data which were obtained on a V-8 experimental engine converted to single-cylinder operation showed a considerable reduction of HC and NO_x (g/ihp-hr) when the intake valve opening was advanced from standard timing. Retarding the intake valve opening slightly increased the NO_x emission but had no effect on hydrocarbons. For the particular test conditions shown intake valve opening timing had no significant effect on CO or specific fuel consumption.

In another series of tests the effect of exhaust valve closing timing was investigated. Figure 4-49 presents the results. Advancing the closing time of the exhaust valve by about 50 degrees resulted in a 25-percent reduction of HC and a 45-percent reduction of NO_x. Retarding the closing time by 35 degrees resulted in HC and NO_x reductions of about 18 and 50 percent, respectively.

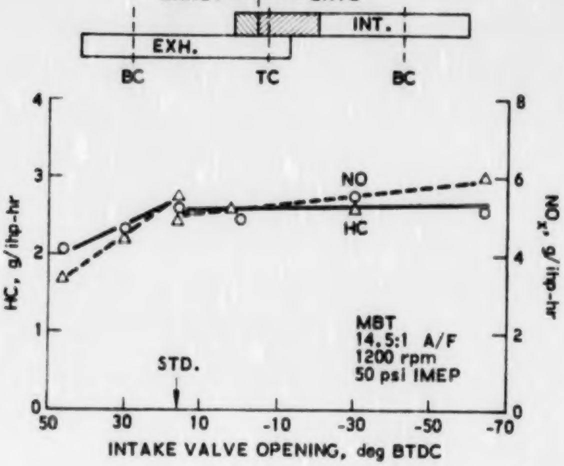


Figure 4-48. Effect of intake valve opening timing on HC and NO emissions (Ref. 4-39)

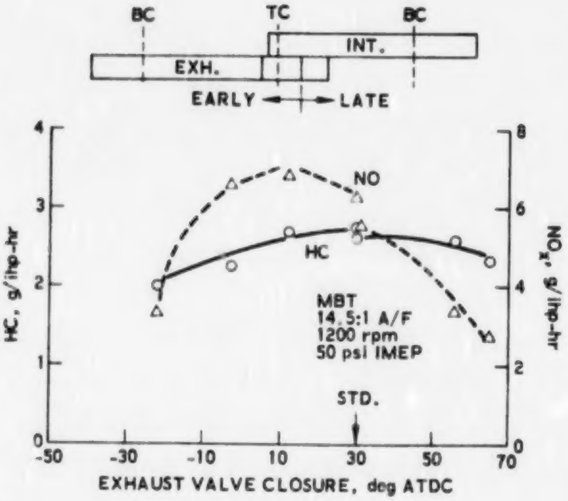


Figure 4-49. Effect of exhaust valve closing on HC and NO emissions (Ref. 4-39)

Apparently early closing of the exhaust valve prevented the discharge of hydrocarbon-rich quench gases (Ref. 4-39) and was therefore more effective in reducing the emission of hydrocarbons than late closing.

On the other hand, in the case of NO_x abatement late closing of the exhaust valve was more effective because of higher cooling of the residual gases in the inlet manifold. Again exhaust valve closing timing had no significant effect on CO and specific fuel consumption.

Valve timing, and other parameters such as exhaust back pressure, air-fuel mixture temperature, engine load, engine speed, air-fuel ratio, spark retard and external exhaust gas recirculation (EGR) singly or in combination have pronounced effects on NO_x. This is illustrated in Figure 4-50 (Ref. 4-47). As indicated NO_x depends only on the amount of charge dilution and is independent of the dilution method used. However, the relationship between spark retard and exhaust back pressure and NO_x is more complex. On the basis of this comparison, it was concluded that not all engine variables exhibited a consistent relationship between NO emission and charge dilution and that charge dilution, although quite important, is definitely not the only factor affecting NO emission.

Emission test results obtained on a car equipped with a 350 CID engine with 9:1 compression ratio are depicted in Figure 4-51, showing the effects of cam advance and retard on NO_x (1970 Federal test procedure). Both advancing and retarding the camshaft by 40 degrees reduced NO_x emission by about 40 to 50 percent. In addition, a small reduction in HC was observed (Ref. 4-48). The idle quality was good, but there was a significant loss in power resulting in poor performance at part throttle. However, at speeds above 1200 rpm driveability was acceptable.

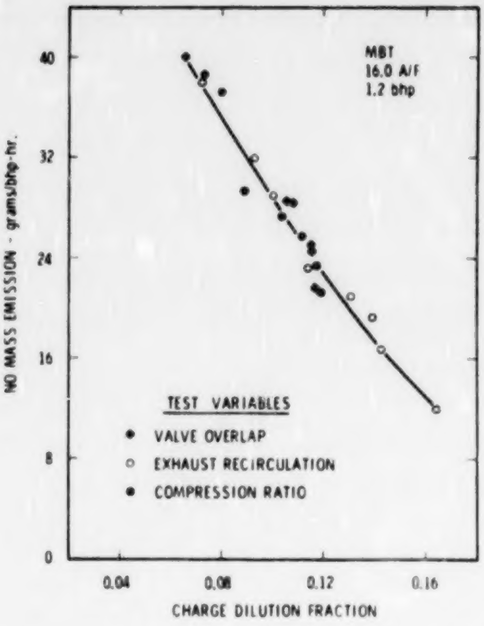


Figure 4-50. Nitric oxide vs charge dilution relationship for valve overlap, recirculation and compression ratio tests (Ref. 4-47)

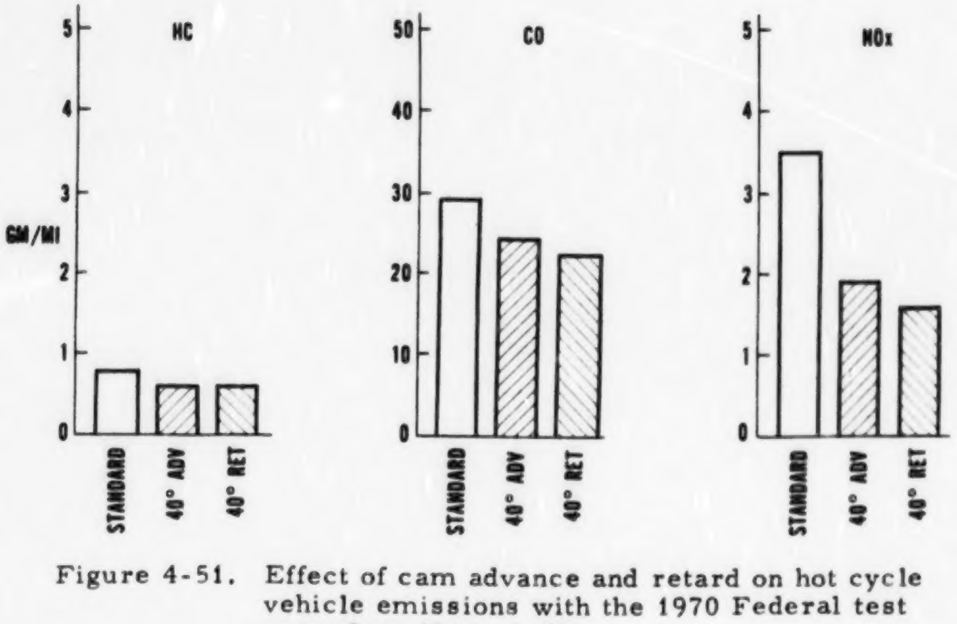


Figure 4-51. Effect of cam advance and retard on hot cycle vehicle emissions with the 1970 Federal test procedure (Ref. 4-48)

In another test utilizing variable valve overlap NO_x was reduced to about 1.2 g/mile, (1970 Federal test procedure) compared to 3.5 g/mile with the standard camshaft. However, HC was higher mainly as a result of the richer idle mixtures required to reduce misfire.

Complementary information on variable cam timing as a tool for emission control is presented in References 4-49 and 4-50.

Tests conducted by Freeman et al (Ref. 4-48) indicate that the fuel consumption increased slightly as the cam was advanced and decreased slightly as the cam was retarded. Advancing the cam causes opening of the exhaust valves before the end of the expansion stroke. Thus the cylinder pressure is relieved prematurely resulting in some loss of power and fuel economy.

4.2.1.7 Engine Load

Generally, the concentration of NO_x increases markedly with increasing inlet manifold air pressure (increasing engine load), as shown in Figure 4-52 (Ref. 4-34). The increase of NO_x with increasing engine load is due to the attendant rise of the combustion temperature and the reduction of the mixture dilution by residual exhaust gases in the engine cylinder (Ref. 4-49).

Engine load also has an effect on the average surface temperature (AST) of the engine cylinder walls and consequently will influence the "wall quenching" phenomenon and the emission of unburned hydrocarbons. Test data indicate that, on the average, only about 43 percent of the decrease in exhaust hydrocarbon, which occurred when indicated power was increased from 2 to 13 hp, can be attributed to increasing AST (Ref. 4-40). Other factors most likely to affect HC as power output is increased are quench zone gas temperature, pressure, turbulence, and after-reactions.

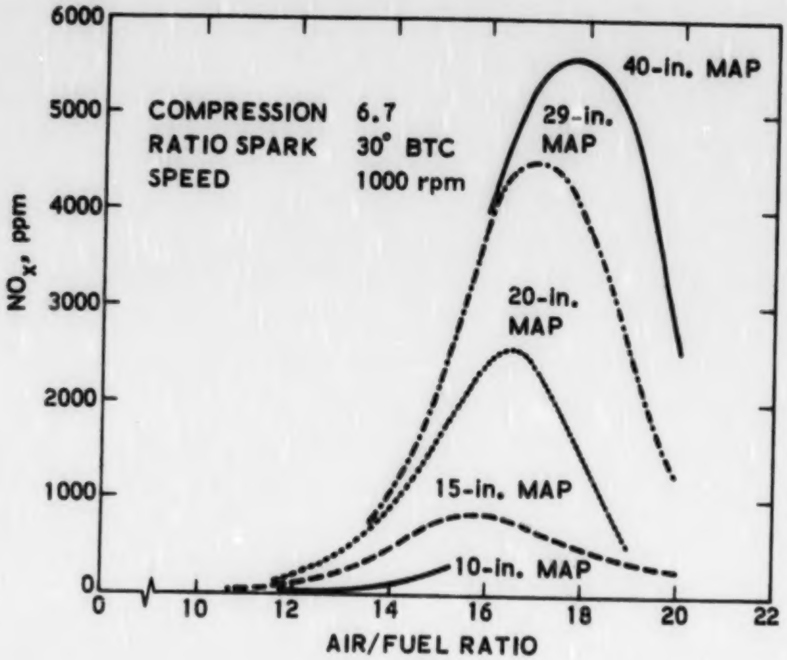


Figure 4-52. Effect of manifold air pressure on oxides of nitrogen (Ref. 4-34)

Since the average surface temperature has been identified as a variable affecting exhaust hydrocarbon concentration, methods of increasing this temperature represent an interesting HC abatement technique (Ref. 4-40). Unfortunately, any increase in AST adversely affects engine octane requirement and volumetric efficiency. The effect of engine load on CO appears to be very small.

Test data from a large stationary two-stroke gas engine are shown in Figure 4-53 (Ref. 4-31). Generally, the same trends of NO_x, HC, and CO, as in the case of automotive spark ignition engines, are apparent. The decline in NO_x as engine load is increased above 95 psi bmep is believed to be due to progressive enrichment of the mixture.

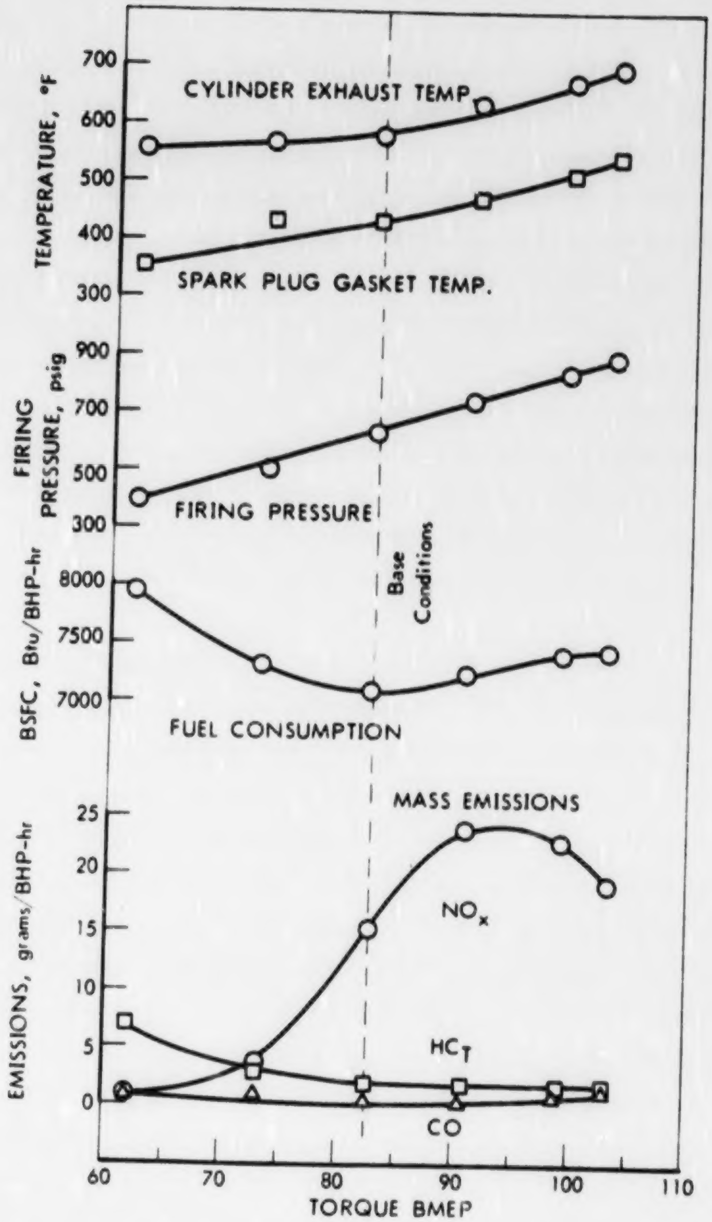


Figure 4-53. Effect of load at constant speed on emissions and performance - large two-stroke atmospheric spark-gas engine, base conditions, 300 rpm (Ref. 4-31)

4.2.1.8 Exhaust Backpressure

Test data published by several investigators indicate that exhaust backpressure can affect the exhaust emissions from spark ignition engines (Ref. 4-46). For example, Eltinge et al (Ref. 4-51) concluded that raising the exhaust backpressure lowers the average hydrocarbon concentration of the exhaust gases, partly because it inhibits the exhaust of the last part of the charge, which has the highest hydrocarbon concentration (Ref. 4-39). Furthermore, increasing backpressure in the exhaust system increases the oxidation reaction potential in the exhaust manifold, particularly when the increased backpressure is accompanied by higher exhaust gas temperature. To minimize fuel consumption losses, a sophisticated exhaust backpressure control system might be employed, which would raise the exhaust backpressure only at certain engine operating conditions.

Test data from a V-8, 351 CID gasoline engine are presented in Figure 4-54 (Ref. 4-52), showing the effect of exhaust backpressure on emissions. Increasing the exhaust pressure from 30 in Hg absolute up to 44 in Hg resulted in a moderate reduction of the NO_x and HC exhaust concentration. Since the exhaust backpressure adversely affects the engine power output, the reduction in HC and NO_x on a specific mass basis (g/bhp-hr) would be even less. However, there are indications that elevated exhaust pressure might be more effective in conjunction with increased valve overlap (Ref. 4-46). The adverse effects of exhaust backpressure on engine power and fuel economy might be counteracted by expanding the exhaust pressure through a turbocharger. In Ref. 4-46, it was concluded that the relative effect

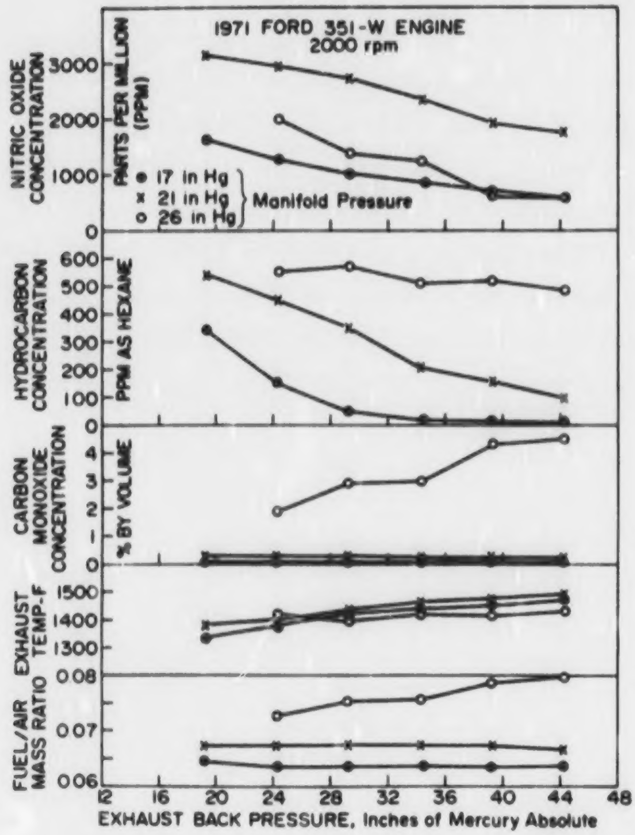


Figure 4-54. Exhaust emissions, exhaust temperature, and fuel-air ratio as functions of exhaust backpressure for three absolute inlet manifold pressures, 2000 rpm (Ref. 4-52)

of increased exhaust backpressure on exhaust emissions was only slight at standard valve timing, but quite noticeable in the case of increased valve overlap.

4.2.1.9 Combustion Chamber Deposits

Extensive laboratory and field test results show that the combustion chamber deposits exhibit a significant effect on the exhaust emissions from spark-ignition internal combustion engines. As an example, Figure 4-55 (Ref. 4-53) presents the effect of deposit buildup on the walls of the engine combustion chamber on HC and NO_x. Before the start of the test, the combustion chamber deposits were removed from all cylinders. As indicated, the emissions of NO and HC increased steadily with the buildup of deposite during the engine operation. After 142 hours of operation, the deposits were removed and the exhaust concentration of nitric oxide dropped nearly 25 percent and the concentration of exhaust hydrocarbons dropped about 66 percent. Similar results were observed in a fleet test program (Ref. 4-54).

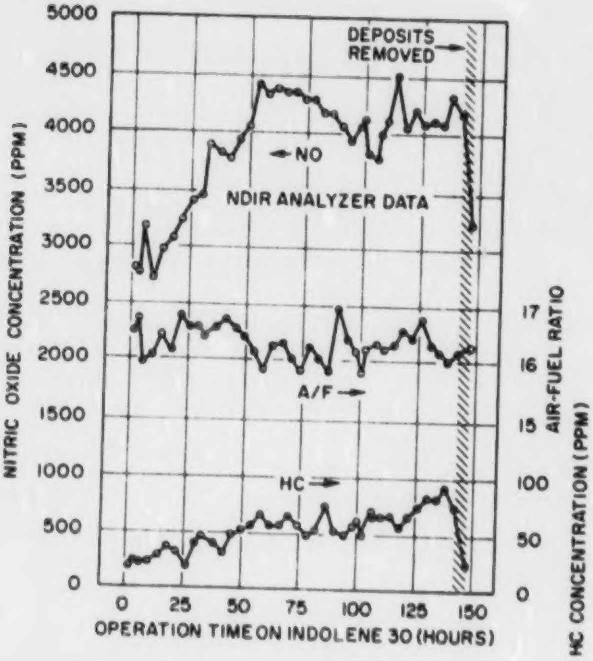


Figure 4-55. Effect of deposit buildup on exhaust NO and HC concentrations (Ref. 4-53)

The effect of leaded and unleaded gasolines on exhaust emissions as influenced by combustion chamber deposits is presented in Ref. 4-55. On the basis of an intensive state-of-the-art review of numerous test data submitted by 18 different companies, it was concluded that (1) lead content in gasoline has a significant effect on the formation of combustion chamber deposits, (2) the equilibrium deposit level relative to hydrocarbon emission is 7 to 20 percent higher when leaded instead of unleaded gasoline is used, (3) the presence of lead in gasoline has no effect on carbon monoxide emission, and (4) no lead content effects on nitric oxide emission have been observed.

An investigation conducted with a number of gasoline additives has indicated that a reduction in HC by approximately 50 percent was possible. This effect is attributed to an attendant modification of the deposits in the engine (Ref. 4-56).

4.2.2 Preventive Emission Control by Engine Modification

4.2.2.1 Combustion Chamber Modification

The design of the combustion chamber of a spark ignition engine has an important influence on its performance, fuel economy, and NO_x and HC emissions (Refs. 4-38 and 4-57). The shape of the combustion chamber and the relative location of the valves affect the flow pattern and the degree of turbulence, and consequently the speed of flame propagation and the thickness of the wall boundary layer. Intensive turbulence tends to decrease the thickness of the wall boundary layer containing unburned fuel and promotes its post-flame oxidation. However, intensive turbulence increases the heat transfer to the cylinder walls and tends to increase the combustion pressure rise rate, the peak combustion temperature, and the formation of NO_x. On the other hand, intensive turbulence tends to decrease the selective retention of hydrocarbon rich boundary layer gases in the combustion chamber during the exhaust cycle, and for this reason may in some cases cause high emission of exhaust hydrocarbons.

The effect of combustion chamber shape and spark plug location on NO_x is shown in Figure 4-56 (Ref. 4-58). As indicated, the chamber having the highest surface-to-volume ratio (configuration No. 3) shows the lowest NO_x emission. Conversely, configuration No. 4 is the most compact chamber, which results in the highest NO_x emission.

The influence of other factors on surface-to-volume (S/V) ratio is shown in Figure 4-57. For example, as the combustion chamber becomes smaller, S/V increases. The effect of S/V on HC is shown in Figure 4-58, depicting test results obtained on a fleet of cars equipped with similar engines (Ref. 4-50). The S/V ratio can be varied by changing the compression ratio of the engine (Ref. 4-59). Based on steady-state dynamometer and vehicle tests, the specific mass emissions of NO_x have been shown to be independent of compression ratio as long as the engine is operated on rich mixtures. However,

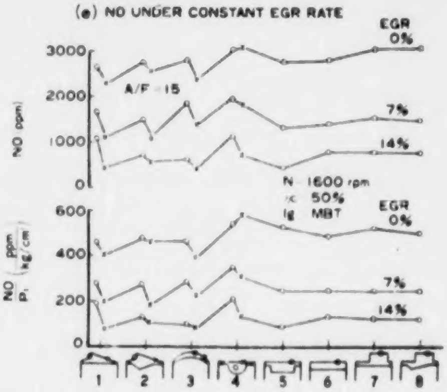


Figure 4-56. NO emission per unit output for different combustion chamber shapes and spark plug locations (Ref. 4-58)

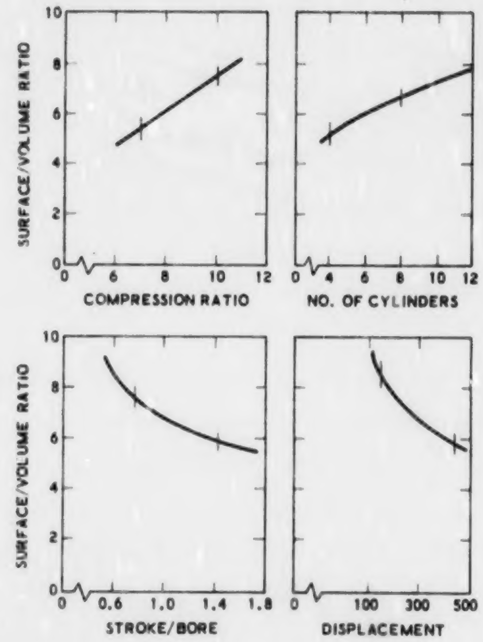


Figure 4-57. Effect of combustion chamber geometry on the surface-to-volume ratio (Ref. 4-50)

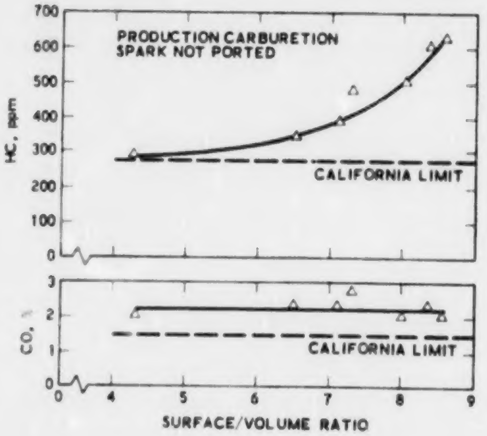


Figure 4-58. Composite values: California chassis dynamometer schedule (Ref. 4-50)

for near-stoichiometric and lean mixtures, NO_x reaches a maximum at compression ratios between 8:1 and 9:1 and decreases for lower and higher compression ratios (Ref. 4-59).

4.2.2.2 Fuel System Modifications

Generally, automotive spark ignition engines are designed for operation on gasoline which is supplied by means of carburetion or fuel injection. However, more recently, a considerable number of light-duty vehicles and trucks have been converted to gaseous fuels (LPG, natural gas).

Since the close of the last century, thousands of patents have been granted for numerous carburetion and fuel injection concepts. The general goal of these inventions was to improve the engine performance and fuel economy and, in some cases, extend the engine operation to heavy or less volitile fuels. Fuel injection is favored by some automobile manufacturers in order to improve the mixture uniformity. As a result, an extension of the lean limit of engine operation can be achieved with a concomitant improvement in fuel economy and emissions. However, there are indications that similar gains might be attained with more sophisticated carburetion systems. For example, a mixture-optimizer system for use in conjunction with a conventional carburetor or fuel-injection system has been recently developed (Ref. 4-60). It includes a feedback-type electronic control device which automatically selects the air-fuel ratio, yielding minimum fuel consumption at all operating points. Apparently, the minimum fuel consumption occurs for mixture ratios close to the misfire limit. Although no test data are available from this system, the emissions are expected to be low.

The lean limit of engine operation can be extended by means of charge homogenation, which might be accomplished by passing the air-fuel mixture generated in a conventional carburetor through a vaporization tank. For example, utilization of a steam-jacketed tank designed to completely vaporize the liquid fuel and thoroughly mix the fuel vapor and inlet air before induction into the engine permitted an extension of the lean limit from an air-fuel ratio of about 17 to about 21. This improvement, which is attributed to improvements in the fuel distribution and cyclic cylinder pressure variations, resulted in substantial reductions in HC and CO (Ref. 4-37).

In a more recent investigation, charge homogenation was accomplished by vaporization of the fuel and subsequent mixing of the vapor with unheated inlet air (Ref. 4-61). In this case, NO_x was reduced considerably, accompanied by some reduction in HC and CO. Another fuel atomizing carburetion system including EGR was tested in an automobile, and substantial HC, CO, and NO_x reductions relative to the standard carburetor were achieved (Ref. 4-36).

4.2.2.3 Inlet Manifold Optimization

The size, shape, and length of the inlet manifold have an important effect on the air-fuel mixture formation, fuel distribution to the individual cylinders, and the resulting flow pattern of the charge in the cylinders. This in turn has an important effect on the engine combustion process (Refs. 4-38, 4-57, 4-62, and 4-63).

In general, the smaller the cross-sectional area of the inlet manifold, the higher the flow velocity in the manifold, resulting in high turbulence and improved mixing of the charge. The attendant loss in volumetric efficiency can be alleviated by means of a dual inlet manifold (Ref. 4-64). In this device, a manifold having a small cross section was fed from a small-bore carburetor while a large manifold was coupled to a conventional two-barrel carburetor. At low engine load, the small manifold supplied the charge, while the large manifold took over at high loads. Although a considerable reduction in HC and CO was obtained with this device, the driveability of the automobile was not satisfactory.

4.2.2.4 Stratified Charge Concepts

The principal objective of charge stratification in spark-ignition engines is the achievement of an extension of the lean limit of engine operation. In this case, the flame front originates in a region of high-energy, near-stoichiometric air-fuel mixture, and propagates from there into lean zones of the chamber.

The charge stratification can be accomplished either by localized fuel injection into a single combustion chamber (open-chamber stratified charge engines) or by supplying a rich mixture to a prechamber, and a lean mixture or pure air to a secondary (main) chamber, which is connected to the prechamber by means of a communicating passage (divided-chamber stratified charge engines).

In the open-chamber configuration, exemplified by the Texaco TCCS (Ref. 4-65) and Ford PROCO (Ref. 4-66) engines, a single combustion chamber is employed similar to that of conventional spark-ignition engines. During engine operation, an air swirl is set up in the cylinder by means of directional intake porting, combined with special piston cup designs. Fuel is injected into each cylinder toward the end of the compression stroke. Upon ignition of the swirling, rich mixture surrounding the spark plug, the burning charge expands into the lower regions of the combustion chamber where the combustion process is then completed in an oxygen-rich environment. This is shown schematically in Figure 4-59. Attempts are currently being made to replace the more expensive fuel injection systems employed in these engines by conventional carburetors.

The divided-chamber stratified charge engines or prechamber engines, exemplified by Honda's CVCC engine concept (Ref. 4-67), employ two interconnected combustion chambers per cylinder. During the compression stroke of the piston, a fuel-rich mixture is inducted into the generally small prechamber while the main chamber is charged with a lean mixture or even pure air. Upon

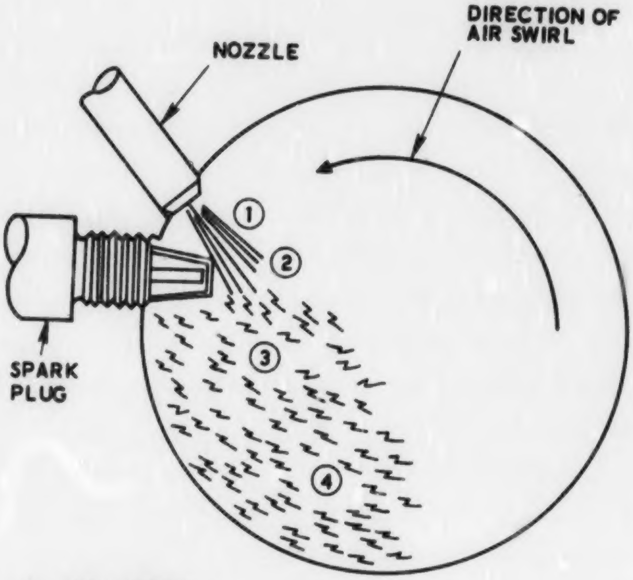

DIRECTION OF AIR SWIRL
NOZZLE
①
②
③
④
SPARK PLUG
1. FUEL SPRAY
2. FUEL - AIR MIXING ZONE
3. FLAME FRONT AREA
4. COMBUSTION PRODUCTS

Figure 4-59. Texaco Controlled Combustion System (Ref. 4-65)

ignition in the prechamber, hot gases expand into the main chamber where combustion is then carried to completion. The principal advantage of prechamber engines over conventional engines is their ability to operate with very lean overall air-fuel mixtures, resulting in low emissions, particularly NO_x. However, because of the less favorable combustion chamber surface-to-volume ratio combined with high turbulence, the heat losses of this engine tend to be higher in conventional designs. The benefits in terms of emission reduction and fuel economy improvement that might be realized in a particular design depend upon the tradeoffs between the heat losses and the inherently higher thermodynamic cycle efficiency obtained with operation in the lean air-fuel mixture regime.

Without incorporation of emission control systems, the HC and NO_x emissions from vehicles adjusted for minimum fuel consumption and equipped with open-chamber stratified charge engines are comparable to those of conventional spark ignition engines, while CO emissions and fuel consumption are substantially lower. Conversely, with emission control, consisting of EGR, oxidation catalysts, and intake air throttling at low power levels, the vehicles meet the statutory 1976 Federal emission standards at low mileage and show equal or slightly better fuel economy than the average 1973 certification vehicles tested.

Both open-chamber and divided-chamber stratified charge engines have demonstrated a lower sensitivity to fuel octane number. Because of the late fuel injection combined with immediate spark ignition, the Texaco TCCS engine can operate on a wide range of diesel fuels. The potential benefits that might be derived from this combustion process include good fuel economy and low NO_x emissions. The stratified charge engines developed to date, notably by Ford, Texaco, and Honda, have achieved an advanced state of development.

4.2.2.5 Exhaust Gas Recirculation

Exhaust gas recirculation (EGR) as a method of NO_x control in internal combustion engines has been under evaluation since the early 1960s (Ref. 4-68). Briefly, the effect of EGR on NO_x is directly related to the attendant reduction in combustion temperature. Since the amount of nitric oxide produced in the engine cylinder is an exponential function of the combustion temperature, even a moderate reduction in the combustion temperature results in a significant decrease in the formation of NO_x.

Experimental and theoretical data relating NO_x reduction to exhaust gas recycle rate are presented in Figure 4-60 for engines operating at conventional air-fuel ratios (Ref. 4-69). The agreement between prediction and test data is good. For low recycle rates,

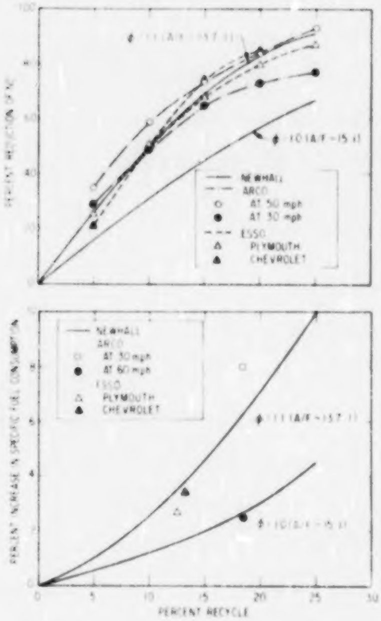


Figure 4-60. Effect of EGR on NO reduction and specific fuel consumption (Ref. 4-69)

the reduction of NO_x is nearly proportional to the amount of exhaust gas recycled. For higher quantities of recycle, the effect diminishes. Substantial (approximately 40 to 80 percent) NO_x reductions are achievable at 10 to 20 percent recycle rates in the conventional air-fuel ratio range. However, because of the dilution of the charge and reduced peak combustion temperature, a reduction in power output occurs (at the same spark advance setting) which effectively translates into a fuel economy loss, as shown in Figure 4-60.

Because of the interrelationship of spark timing, cycle temperature, and power output, it is possible to advance spark timing to avoid or minimize the effects of EGR on power and SFC. In tests performed by Esso (Ref. 4-70), EGR was shown to have a much lower fuel economy penalty than spark retard for the same NO_x reduction. It was found possible to operate with both recycle and some spark advance and obtain some NO_x reduction with a slight improvement (approximately 2 percent) in SFC in one case.

For any given engine, then, the fuel consumption penalty would be strongly influenced by the baseline engine air-fuel ratio and NO_x emission characteristics, the amount of NO_x reduction required to meet a given standard, and the potential for optimizing spark timing and recycle rate within these constraints.

More recently, a number of EGR systems have been subjected to extensive testing and evaluation (Refs. 4-71 to 4-76). Typical test data from one design are presented in Figure 4-61 (Ref. 4-75), showing NO_x (in g/mile) and fuel economy obtained on a 318 CID, V-8 Plymouth engine, as a function of air-fuel ratio and EGR flow rate. It is of interest to note that in this case, a small improvement in fuel economy was realized by operating the engine with 4 to 9 percent EGR and an air-fuel ratio between 16 and 18. Test data from another study indicate that some improvement in fuel economy might be achieved with EGR, particularly when spark timing is advanced to maximum power (Ref 4-77).

There are indications that the use of EGR might result in a reduction of piston ring wear (Ref. 4-78). In one engine, incorporation of 12 percent EGR resulted in near-zero wear rate after six hours of steady-state engine operation.

The major concern arising from the use of EGR is related to the occurrence of unstable engine operation, commonly called "power surging." This phenomenon is attributed to an increase in the cyclic pressure variation occurring in the engine cylinder during operation with EGR and the attendant reduction of the lean limit of engine operation. Mixture homogenation appears to be a feasible approach to circumvent this problem.

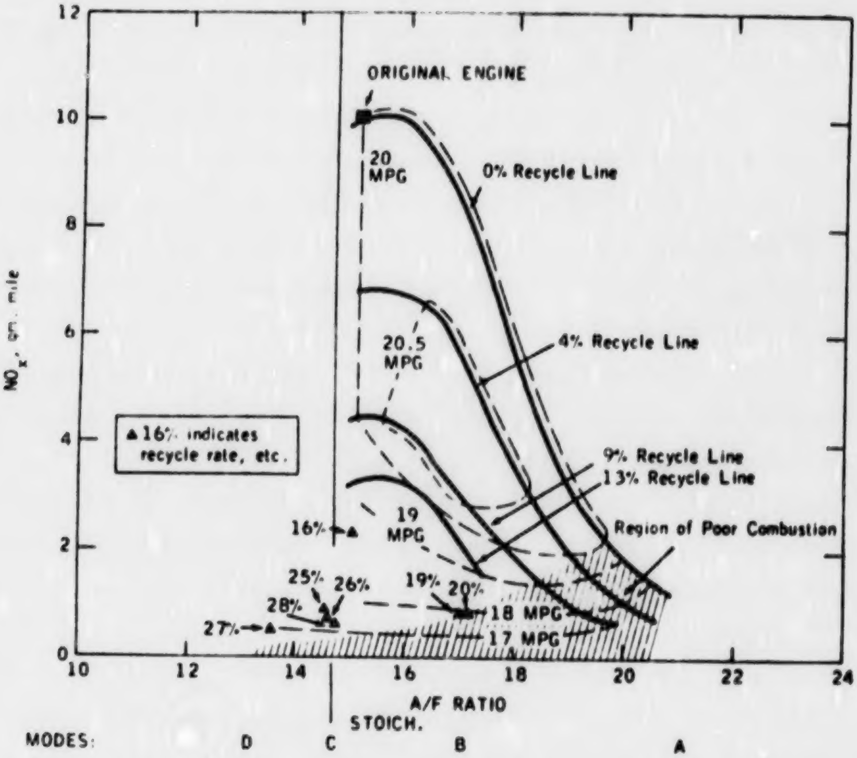


Figure 4-61. Test stand NO_x emissions as a function of A/F and recycle rate - 50 mph road load, 37-degree btdc spark timing, gasoline fuel (Ref. 4-75)

4.2.2.6 Water Injection

Water injection has been used since the early years of kerosene engines as a means of suppressing "engine knock." More recently, water injection has been considered to reduce the NO_x emissions from internal combustion engines. The observed reduction in NO_x is due partly to the latent heat of water and partly due to a change in the specific heat of the mixture. Test data from a single-cylinder CFR engine operated at constant speed and load are presented in Figure 4-62 (Ref. 4-79). In these tests, the loss of power due to water injection was compensated for by increasing the air-fuel ratio of the charge. As indicated, water injection at a rate of one pound of water per one pound of fuel decreased NO_x by about 80 percent, and improved the specific fuel consumption by about two percent.

Based on another study (Ref. 4-80), it appears that water injection may be accompanied by an increase in the volumetric efficiency of the engine due to evaporative cooling of the inlet charge during the induction.

Water injection directly into the cylinder has been shown to increase the effectiveness of the water, particularly when the water is injected during the intake stroke. This is illustrated in Figure 4-63, showing that only 0.6 pound of water per one pound of fuel was required to obtain an 80 percent reduction in NO_x (Ref. 4-81). This improvement is accompanied by a slight increase in HC. Similar results have been achieved over a wide range of air-fuel ratios.

In the past, attempts have been made to simplify the water injection procedure by using water-fuel emulsions. Limited test data indicate that the reduction in NO_x is about the same as in the case of water injection into the inlet manifold.

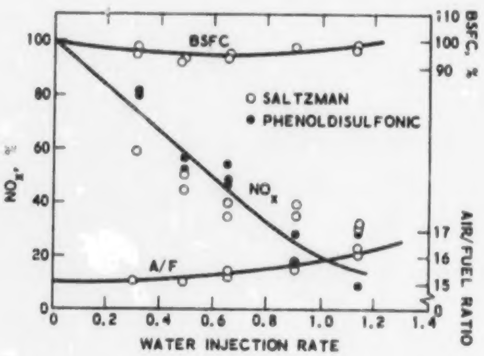


Figure 4-62. Effect of water injection on the emissions and specific fuel consumption of a CFR engine; 5.5 hp, 1200 rpm, 30° spark advance (Ref. 4-79)

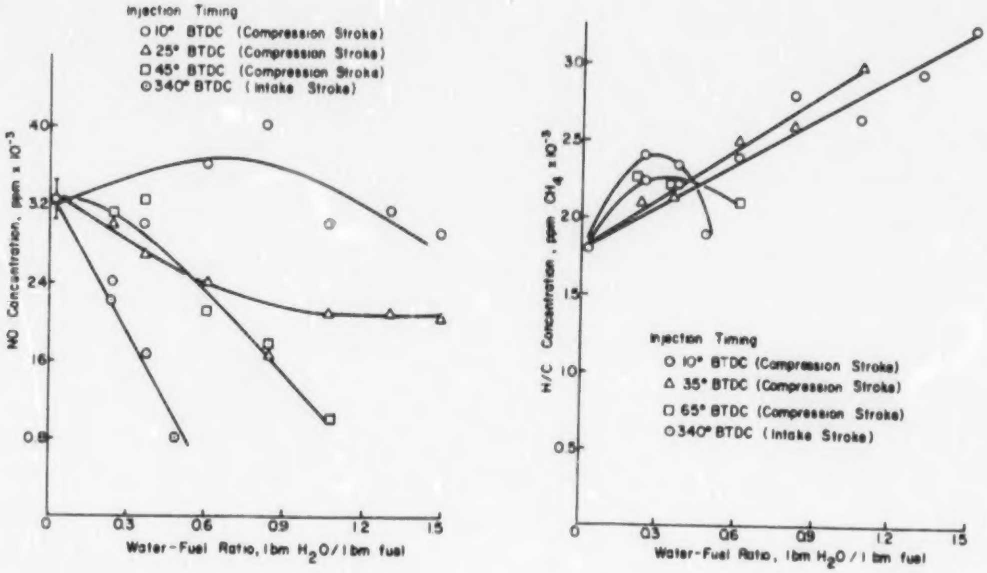


Figure 4-63. Effect of water injection on NO and HC concentration, 900 rpm; spark advance 30° btdc (Ref. 4-81)

Water injection test data from an Ingersoll-Rand PKVGR-12, four-stroke, naturally aspirated stationary gas engine are presented in Figure 4-64 (Ref. 4-82). The trends of NO_x and HC mass emissions, as affected by water injection rate, are similar to the emission trends obtained in automotive spark ignition engines. However, the specific fuel consumption increases rapidly with increasing water flow rate. The increase of CO shown in the figure may be due partly to the reduction of the air-fuel ratio and partly to flame quenching caused by non-uniformly distributed water spray.

4.2.3 Fuel Modification

The prospect of spark ignition automotive engine exhaust emission control by means of fuel modifications has been the target of

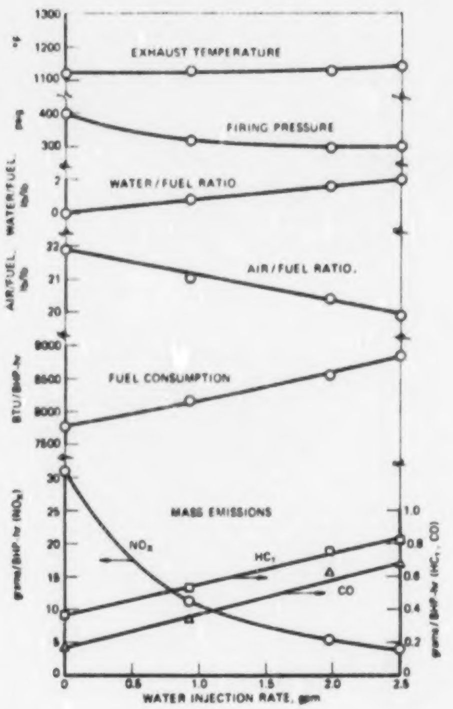


Figure 4-64. Effect of water injection on emissions and performance - Ingersoll-Rand PKVGR-12, 4-cycle naturally aspirated spark-gas engine (Ref. 4-82)

numerous investigations for over two decades (Ref. 4-83). Most of the earlier research was aimed at the investigation of the smog-forming potential of commercial gasoline blends. More recently, attention has been directed to the effects of fuel volatility on engine emissions (Ref. 4-84). In addition to the effect on evaporative losses from the carburetor and fuel tank, the volatility of gasoline blends has an effect on the reactivity index of the exhaust hydrocarbons. However, no significant effect on nitric oxide and carbon monoxide emission has been found.

Exhaust emission of polynuclear aromatic hydrocarbons (PNA) and of phenols has been studied with a variety of test fuels, using cyclic tests in five vehicles including one without emission control, two with engine modification control, and two with experimental very-low-emission systems (Ref. 4-85). The tests revealed that the fuel composition influenced emissions both directly and through buildup of engine deposits.

NO_x and CO emitted from spark ignition engines can be influenced to some extent by fuel composition (Refs. 4-86 and 4-87). Carbon-to-hydrogen ratio of the fuel appears to be equally influential as the energy content of the fuel in determining the relative production of nitric oxide and carbon monoxide (Refs. 4-87 and 4-88).

Partial or complete substitution of gasoline by unconventional fuels (e.g., methanol, ethanol, methane, propane, hydrogen, ammonia, hydrazine, etc.) is currently being investigated from the point of view of exhaust emission reduction as well as economic feasibility (Ref. 4-89). Some fuels (e.g., methane and hydrogen) offer the possibility of engine operation essentially free of exhaust hydrocarbons and strongly reduced nitric oxide emission.

Relatively little information has been published on the effects of fuel additives on the exhaust emissions from spark ignition engines. Reference 4-90 presents limited experimental data of the effect of non-metallic combustible additives and metallic compounds on NO production. Another investigation was concerned with gasoline additives which form a low surface-tension coating in the engine induction system, As a result, an improved fuel distribution has been achieved, accompanied by improvements in driveability, fuel economy (3 to 4 percent) and HC emissions (15 to 25 percent) (Ref. 4-91).

4.2.4 Corrective Emission Control

A considerable amount of work has been conducted to date on corrective emission control approaches, including thermal reactors and catalytic converters. These are briefly discussed in the following sections.

4.2.4.1 Thermal Reactors (Lean and Rich)

A thermal reactor is a chamber (replacing the conventional engine exhaust manifold) into which the hot exhaust gases from the engine are passed. The chamber is sized and configured to increase the residence time of the gases and permit further chemical reactions, thus reducing HC and CO concentrations. In general, the thermal reactor embodies a double-walled and insulated (between walls) configuration, with port liners to direct the exhaust gases to its inner core section. In some instances, baffles and/or swirl plates are used to further promote mixing.

There are two different types of thermal reactors in research and development by several companies: the rich thermal reactor (RTR) and the lean thermal reactor (LTR). The RTR is designed for fuel-rich engine operation. As a result of the chemically-reducing atmosphere and lower combustion temperatures in the engine combustion chamber, the amount of NO_x formed in the engine is reduced. If the engine is run sufficiently rich (air-fuel ratio approximately 11-12), it is possible to limit the formation of NO_x to less than 2 g/mi; however, fuel economy penalties at these rich mixtures are as high as 20 percent. As the exhaust from the cylinders contains large quantities of HC and CO, secondary air supplied by a pump is injected into the reactor to permit further oxidation of these species.

The thermal reactor should be designed for minimum thermal capacity to minimize cold-start emissions. Since relatively high temperatures (1700 - 2000°F) are achieved in the RTR, high-temperature materials (e.g., Inconel 601 containing 60-percent Ni, 23-percent Cr, 14-percent Fe, 1-1/2-percent Al) are required for the inner core, baffles, and port liners. At these high temperatures, engine misfiring, which produces high HC levels, could lead to excessive local temperatures and material burnout conditions in the RTR;

therefore, temperature control devices are necessary to protect it. Ceramic materials, which could be more tolerant to overtemperature conditions than metals, have not demonstrated to date the necessary thermal and mechanical shock properties. A summary description of proposed experimental designs for RTR types is presented in Table 4-9 (Ref. 4-70).

As mentioned previously, a carburetor calibration change of three air-fuel units (15 to 12) to minimize NO_x to less than 2 gm/mi may incur a fuel economy penalty of 15 to 20 percent. According to Figure 4-65, if an exhaust gas recirculation system is added to the RTR to further control NO_x to levels below approximately 1 g/mi, fuel economy penalties are as high as 20 to 30 percent (Ref. 4-69).

TABLE 4-9. THERMAL REACTOR SUMMARY (Ref. 4-70)

Reactor Type	Induction Air-Fuel Ratio	Reactor Operating Temperature	Reactor Volume, ($in.^3$)		Reactor Air Injection	Port Liners
			Core	Total		
Rich Reactors						
Du Pont Type V	14	(a)	60	(a)	Yes	Yes
Du Pont Type VII(b)	11.5 - 12.5	(a)	(a)	(a)	Yes	Yes
Esso Synchrothermal	12.2	1600 - 1900(c)	103	(a)	Yes	No
Esso Modified RAM	11 - 13	1600 - 1750	71[d]	(a)	Yes	No(e)
IIEC/Ford Type H	(a)	1600 - 1850(cf)	40	97	Yes	Yes
IIEC/Toyo Kogyo	(a)	1600 - 1800(cf)	61	(a)	Yes	Yes
IIEC/Nissan	(a)	(a)	(a)	220	Yes	Yes
British Small Engine	10 - 14	1600 - 1650	70	(a)	Yes	No
Lean Reactor						
Ethyl Lean Reactor	17 - 19	1400 - 1600(c)	-	160	No(g)	Yes

[a] Not specified

[b] Recent Du Pont system

[c] With EGR

[d] Torus volume

[e] Flameholders

[f] Thermal protection cutoff temperature

[g] Air injection possibly required during choke period

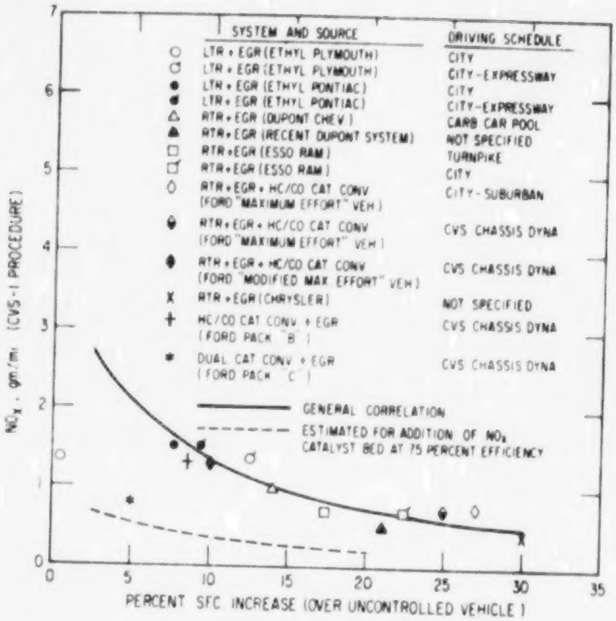


Figure 4-65. NO_x versus SFC increase (Ref. 4-69)

The LTR is used in conjunction with an engine operated on the lean side of stoichiometric mixtures; i.e., with excess air. Currently LTR systems are limited to air-fuel ratios of approximately 19. In this case, the HC and CO emissions are much lower than in the case of the RTR (but NO_x levels are somewhat higher). Therefore, little chemical heat is generated in the reactor and its temperature is governed to a large extent by the sensible heat in the exhaust gas. This means that the oxidation of HC and CO is accomplished within the LTR at lower temperatures than for the RTR, and without the requirement for additional air (i.e., no air pump is needed). Because of the lower operating temperatures, the durability requirement can be met by less expensive materials for the construction of the reactor core and baffles; however, careful attention must be given to minimizing heat losses or the conversion is limited by low reaction rates. On the other hand, more stringent requirements exist for engine air-fuel mixture control and cylinder-to-cylinder fuel distribution. This may require utilization of an advanced carburetor or electronic fuel injection. EGR is generally added for additional NO_x control. Although little or no fuel economy penalty is chargeable to the LTR itself, with EGR an approximate 10-percent decrease in fuel economy is realized for NO_x levels of approximately 1.5 g/mi. Peak power loss due to lean operation causes a small loss in vehicle performance.

The Ethyl Corporation lean reactor is the only known design of a lean operating system for which specific details of configuration and performance are available. It is designed for operation at air-fuel ratios of between 17 and 19. As shown in Table 4-9, its operating temperature is 1400 to 1600°F, or 200 to 300 degrees lower than those for rich reactor systems. The reactor is cylindrical and consists of an open-tube liner made of 310 stainless steel, surrounded by a layer of insulation which in turn is enclosed by an outer casing of 310 or 430 sheet stainless steel (Ref. 4-69).

4.2.4.2 Catalytic Converters (Oxidizing and Reducing)

The oxidation of the HC and CO in the exhaust gas can be accomplished by means of a catalyst at temperatures lower than those in a thermal reactor. A catalytic converter can be placed farther from the engine than a thermal reactor and can maintain its effectiveness without rich mixture engine operation (normally required to maintain the necessary chemical energy level of the exhaust gases in a thermal reactor). As a consequence, the fuel economy penalty is lower. Generally, secondary air for oxidation has to be supplied similarly, as for the thermal reactors; however, the control of the secondary air flow rate is critical, because overheating due to excessive oxidation reactions in the converter can destroy the effectiveness of the catalyst. For the same reason, the carburetor of the spark ignition engine is required to maintain accurately the predetermined mixture air-fuel ratio, under all engine operating conditions (Ref. 4-92).

The need for more close control of the exhaust-gas composition in the catalytic converters led to a closed-loop system of fuel metering. In such a system, an oxygen sensor is used to detect the level of oxygen in the exhaust stream and to supply an error feedback signal either to an electronic fuel-injection control module or to a specially constructed carburetor (Ref. 4-93). This signal causes adjustment in the fuel or air supply, thereby maintaining close control of air-fuel ratio.

The automotive emission control systems in development to meet the statutory 1976 emission standards utilize two catalyst beds to reduce NO_x as well as HC and CO emissions. The bed closest to the engine is used to remove NO_x and is operated in a reducing environment. Secondary air is then added to the exhaust stream between the catalyst beds (downstream of the NO_x reducing catalyst bed), and the remaining HC and CO are removed in the second catalyst, the oxidation bed. At least one catalyst manufacturer has been working on the development of a tricomponent catalyst (single bed) which, under carefully controlled operating conditions, simultaneously promotes the oxidation of HC and CO, and the reduction of NO_x. Examples of best low-mileage emissions obtained with dual catalyst systems are presented in Table 4-10 (Ref. 4-94).

Lead additives, as well as sulfur and phosphorus content, are toxic to catalyst materials, resulting in a degradation of its effectiveness (Ref. 4-69). This is illustrated in Table 4-11, which summarizes the results of the most promising durability tests conducted on some of the dual catalyst vehicles (Ref. 4-94). The causes of the rapid deterioration in NO_x catalyst efficiency are not yet quantitatively understood. However, it is clear that poisoning of the active catalyst material by contaminants in the fuel (nonleaded gasolines) is the cause of part of the observed deterioration.

TABLE 4-10. EXAMPLES OF BEST LOW MILEAGE EMISSION MEASUREMENTS WITH DUAL-CATALYST SYSTEMS ON EXPERIMENTAL 1976 VEHICLES (Ref. 4-94)

Company	Vehicle Weight, lb	Engine Size, CID	EGR	Emissions, g/mi[a]			Catalyst Data
				HC	CO	NO_x	(1) $\frac{HC}{CO}$ (2) NO_x
American Motors		258		0.27	5.7	0.55	
Chrysler		360	No	0.17	3.0	0.5	(1) (2) Noble, monolith
		360	No	0.23	2.5	0.52	(1) (2) Base
		360	Yes	0.34	3.9	0.44	(1) (2) Noble, pellets
General Motors	4500	350	Yes	0.24	1.7	0.15	(1) UOP, platinum, pellet (2) Gulf, monolith
	4500	350	Yes	0.42	3.1	0.21	(1) Air products, pellet (2) Gulf, pellet
	4500	350	Yes	0.17	1.0	0.19	(1) UOP, platinum, pellet (2) Johnson-Matthey, monolith
	4500	350	Yes	0.21	1.0	0.22	(1) UOP, platinum, pellet (2) Johnson-Matthey, monolith
	4500	350	Yes	0.37	1.8	0.27	(1) UOP, platinum, pellet (2) GM, pellet
Ford	4000	250	No	0.45	2.9	0.38	(1) Engelhard, monolith (2) Promoted Base, pellet
	5000	351	No	0.43	2.4	0.27	(1) Engelhard, monolith (2) Gould, GEM
	4000	250	No	0.52	3.7	0.39	(1) Engelhard, monolith (2) ICI, pellet
	5000	351	Yes	0.48	3.3	0.39	(1) Engelhard, monolith (2) ICI, pellet

[a] 1975 CVS-CH test procedure. Data were usually averages of several tests and were received up to November 1972.

[b] Emission-control systems include a manifold thermal reactor before the NO_x-reduction catalyst

4.2.5 Control of Emission from Blowby, Carburetor, and Fuel Tank

Positive crankcase ventilation systems have been used in automotive engines since 1962, and are designed to prevent the

TABLE 4-11. EMISSIONS AS FUNCTION OF MILEAGE FOR DURABILITY TESTS ON DUAL-CATALYST SYSTEMS (Ref. 4-94)

Manufacturer, Vehicle[a]	Catalysts	Mileage	Emissions, g/mi[b]			NO_x Catalyst Efficiency,[c] %
	(1) $\frac{HC}{CO}$ (2) NO_x		HC	CO	NO_x	
General Motors						
4500 lb	(1) UOP, noble,	0	--	--	0.22	78[d]
350 CID	pellet	1,000	0.32	1.7	0.42	58[d]
Chevrolet	(2) Gulf, noble,	7,000	0.39	3.0	0.45	55[d]
EGR	pellet	13,000	0.52	4.8	0.73	27[d]
4500 lb	(1) UOP, noble,	0	0.21	1.0	0.21	79[d]
350 CID	pellet	7,000	0.47	1.8	0.59	41[d]
Chevrolet	(2) Johnson-Matthey,					
EGR	noble, monolith					
4500 lb	(1) UOP, noble,	0	0.36	1.8	0.28	72[d]
350 CID	pellet	4,000	0.57	4.1	0.51	49[d]
Chevrolet	General Motors					
EGR	Research, pellet					
Ford						
5000 lb	(1) Engelhard, monolith	Low	0.3	1.5	0.56	78
351 CID		3,000	0.33	1.5	0.49	80
Ford	(2) monolith	6,000	0.48	2.6	0.70	71
EGR		9,000	0.72	1.9	0.89	63
	8 - 10 g of	12,000	0.66	3.6	0.75	64
	platinum,[e] dual-	16,000	0.66	5.4	1.3	46
	bed converter	20,000	0.82	3.8	1.5	37
5000 lb	(1) pellet	Low	0.35	3.8	0.68	70
351 CID	(2) pellet	1,000	0.61	3.3	0.99	-
Ford		2,000	0.59	3.6	1.25	-
EGR	Dual-bed	6,000	0.68	4.2	1.72[f]	25
	converter					
4000 lb	(1) -	Low	0.52	3.7	0.39	89
250 CID	(2) ICI, pellet	4,000	0.65	5.2	0.48	86
Ford						
No EGR						

[a] Emission Control System includes engine modifications, air pump, NO_x catalytic converter, oxidation catalytic converter, and EGR and manifold reactor where noted.

[b] 1975 CVS-CH test procedure. Data received up to November 1972.

[c] NO_x catalyst efficiency is percent NO_x removed in catalytic converter.

[d] NO_x catalyst efficiency estimated from approximate engine NO_x emission of 1 g/mi.

[e] Catalyst judged by vendor not to be available in commercial quantities.

[f] EGR system failure.

escape of blowby gases from the crankcase to the atmosphere by recycling them back to the inlet manifold (Ref. 4-95). Although these systems offer almost 100 percent control of pollutants from the crankcase, they affect the engine exhaust emissions and the engine performance, primarily during engine idling and off-idle operation. The essential part of these systems is a spring-loaded control valve which is subject to clogging and malfunctioning, which may adversely affect the engine performance and exhaust emission (Ref. 4-96). Resulting deposits in the carburetor and increased crankcase corrosion can be minimized by detergent fuels, high-quality lubricants, and frequent servicing of the system.

The design and location of carburetor vents and the volatility of the fuel have a pronounced effect on the evaporative HC emissions from carburetors (Ref. 4-97). The reactivity or "smog-forming potential" of the evaporative HC emission is strongly affected by the composition of the gasoline and decreases with increasing volatility of the initial fuel (Ref. 4-98). In current automobiles, evaporative losses from the carburetor and fuel tank are controlled by means of an adsorption-desorption device which is periodically stripped by purging the canister with air aspirated into the inlet manifold during predetermined operating modes of the engine (Ref. 4-99).

4.2.6 Combined Emission Control Techniques

Based on the above review of preventive and corrective emission control devices/techniques, it is apparent that a considerable number of effective methods and devices have been developed to date for use in automotive spark ignition engines. It is conceivable that even more effective systems might be possible by optimizing the design and operation of the various system components. For example, a system consisting of EGR, lean-mixture carburetion (about 25 percent excess of air), and thermal reactor may prove to be a very cost effective NO_x abatement approach.

4.3 GAS TURBINES

The fundamental mechanism of the formation of the various emission species discussed in Section 3.3.3.2.2 is the same irrespective of the type of combustor and/or cycle. The emission quantities produced and their proportions will vary as a function of the combustor design, operating conditions, fuel composition, etc.

In the following sections a number of emission control devices intended for use in automotive engines and steam boilers will be briefly reviewed and the possible application of these to stationary gas turbines will be discussed in detail.

4.3.1 Automotive Engine Emission Control

The development of a low emission automotive power plant under EPA guidance has proceeded in two directions (Refs. 4-100 and 4-101): one is the emission improvement in the standard spark-ignition engine which now dominates and will probably dominate the automotive field for the next decade; and the other is the work on Advanced Automotive Power Systems (AAPS) for the post-1980 era. The latter is currently centered around the Brayton cycle and the Rankine cycle.

The automotive application of the Brayton cycle consists typically of a regenerative gas turbine with a free power turbine as a drive unit. The AAPS work on gas turbine emissions consists mainly on the development of an improved combustor by means of gas recirculation (Solar, jet-induced circulation) and precise and fine liquid atomization and fuel-air mixing (Aerojet platelet injector); and by two surface combustion concepts represented by porous plate (GE) and catalytic combustor (EPA/NASA program) concepts (Ref. 4-102).

Briefly in the jet-induced circulation concept, impingement of the primary air and fuel streams generates a recirculation pattern of partially burned gas in the primary zone which permits operation at lean equivalence ratios (< 0.5), combined with good fuel atomization by air assistance and lower temperature rise in the primary zone. All these factors will have a beneficial effect on the NO_x formation.

The platelet injector permits formation of a pre-mixed fuel-air charge with a short residence time in the primary combustion zone which was shown to be an important factor in NO_x reduction.

Surface combustion involves the promotion of gas-phase oxidation and reduction reactions between fuel and air in close proximity to a solid surface (Ref. 4-103). Low emission characteristics of this concept, in particular NO_x, result from the combustion process occurring at reduced temperatures. The surface combustion can be either catalytic or noncatalytic. In the noncatalytic surface combustor, such as the porous combustor, a fraction of the heat of combustion is immediately transferred from the flame layer to the adjacent solid surface from which it is extracted by cooling means resulting in combustion temperatures below the adiabatic flame temperature. In the catalytic surface combustor, reduced flame temperatures are achieved by operation with very lean fuel-air mixtures. The catalyst serves the function of promoting chemical reactions which, under these particular operating conditions, would occur too slowly for efficient low-emission burning. In the surface combustor, the fuel must be prevaporized and premixed with air before it is fed through the porous or catalytic surface, and that limits the application of surface combustor devices to fuel gas or light distillates (Ref. 4-103).

All the emission control devices described above are in various phases of development, those for spark ignition engines being most advanced. Important design and operational details of these concepts are presented in Section 4.3.3.3.

4.3.2 Stationary Sources Emission Control

The coal fired utility boilers have been found to be the top ranking sources of NO_x emissions, accounting for 30 percent of the total stationary sources (Ref. 4-104). Gas and oil fired utility boilers contributed 5 and 4 percent, respectively. Consequently, the combustor in utility boilers, primarily coal burners, became the priority subject of EPA-directed programs initiated in 1970 for the development of emission controls. The program consisted of four major components: field testing and surveys; process research and development; fuels research and development; and fundamental combustion research (Ref. 4-104).

Two parts of this program are of particular interest to stationary gas turbines: the fuel research and combustion research.

Part of the fuel research work is focused on the development of burner-combustor combinations which would result in low emissions with fuel oil and gas (Rocketdyne, Dynamic Science). The combustion research is grouped into the chemistry of pollutant formation (Esso, Rocketdyne, BMI); the aerodynamics and physical factors affecting pollutant formation (MIT, United Aircraft, JPL); and mathematical modeling of pollutant formation (Dynamic Science, UARL, Princeton University). This work will include a study of the chemical mechanics of fuel-bound nitrogen conversion to NO_x and a study on the effects of flame quenching and recirculation zones on NO_x reduction (Ref. 4-104).

A number of emission control techniques have been successfully incorporated into several stationary boilers. For example, operation at low excess air (LEA) has resulted in NO_x reductions of up to 35 percent. In a horizontally fired boiler the NO_x was reduced from approximately 700 ppm to 450 ppm by reducing the oxygen level in the fuel gas (decrease in excess air) from 3.5 percent to just over two percent (Ref. 4-105).

Staged combustion or two-stage combustion consists of operating the first stage at fuel-rich mixtures and injecting secondary air into the second stage to complete the combustion process. In the first stage, the NO formation is limited by the unavailability of oxygen. Removal of heat between stages reduces the combustion temperature in the second stage thereby kinetically limiting formation of the NO (Ref. 4-105). Two-stage combustion offers about a 50-percent reduction in NO_x and when combined with LEA reductions up to 90 percent were observed.

Flue gas recirculation to the combustion zone has the principal effect of lowering the peak flame temperature. The oxygen concentration is also lowered and both effects favor the reduction of NO_x. Ref. 4-104 reports a 70-percent reduction of NO_x with gas and 50 percent with fuel oil.

Water or steam injection are other potential methods for NO_x reduction. These techniques are not considered to be satisfactory for steam boilers because of the attendent loss in thermal efficiency (5 to 6 percent) and the cost increase associated with steam or water injection (Refs. 4-104 and 4-105).

Flue gas treatment, an alternate method of NO_x reduction, consists of removal of nitrogen oxides as well as sulfur oxides by either catalytic decomposition or reduction, or adsorption by solids such as metal oxides. Principal deterrents to the use of this method are the cost and longevity of the various catalysts under the high temperature conditions (Ref. 4-105).

Surface combustion has been considered for possible application in large utility boilers (Ref. 4-103). However, the prospects for this concept did not appear bright, even if suitable long life catalysts could be developed, because of the complexity in the design and the large surface areas required for combustion.

4.3.3 Low Emission Combustors for Stationary Gas Turbines

The discussion of the gas turbine emissions presented in Section 3.3.3 led to the conclusion that the emissions which are mainly influenced by the combustor design are NO_x and CO. The formation of HC is closely related to the formation of CO. Smoke composed primarily of unburnt carbon particles is related to HC, and the same applies largely to particulates. Thus, a low CO combustor is generally expected to have low HC and smoke emissions. The SO_2 content is a function of the fuel composition and is almost independent of the combustor design. Consequently, the following discussion shall be limited to NO_x and CO with the tacit understanding that a reduction of those species will automatically result in a reduction of all the other pollutants (except SO_2).

The emission data presented in Section 3.3.3 and the various emission limits, existing or proposed, are summarized in Figure 4-66, in terms of specie concentration. It can be seen that only a very small fraction of the current gas turbines could meet the proposed EPA limits for CO and NO_x, or the Rule 68 limits for NO_x at part load without additional emission control devices such as, for instance, water injection. The situation is even worse for the Rule 67 limits which become progressively more stringent (when expressed in NO_x ppm) for larger plants. Thus, there is indeed a need for the development of emission control devices for stationary gas turbines and they will be discussed below.

4.3.3.1 Design Approach

Since the NO_x and CO are formed during and immediately following the period of the combustion process taking place in the primary zone of the gas turbine combustor, it is the design of this zone and the fuel preparation on which the attention must be focused. To lower the temperature of the primary zone, the combustion must take place at

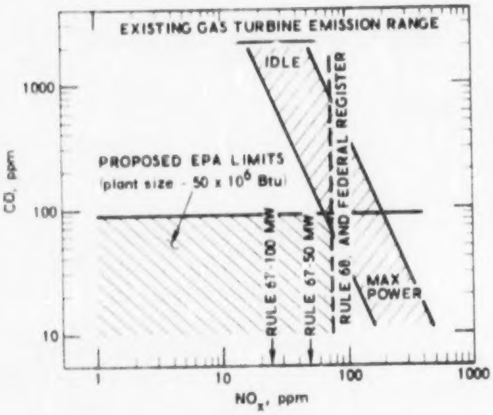


Figure 4-66. Gas turbine state-of-the-art emissions (No. 2 GT turbine oil; 15 percent 0_2)

fuel-air equivalence ratios either well above or below unity. Operating at fuel-rich conditions would require subsequent cooling of the gases to avoid high temperatures in the "afterburning" period. It would also be conducive to formation of CO and it would require greater combustor volume. Consequently, the leaner than stoichiometric fuel-air ratio approach in the primary zone is usually selected by most researchers, although the opposite is true for low emission steam boilers.

Figure 4-67 illustrates the reduction of NO_x as a function of combustor inlet temperature (T_{in}) and fuel-air ratio which jointly determine (for homogeneous mixtures) the flame temperature. The range of T_{in} = 500 to 800° F will correspond to a simple cycle, while T_{in} = 800 to 1500° F will correspond to a regenerative cycle. Fortunately, in the actual combustion process the quantity of NO is kinetically limited because of the short residence time of the combustion gases, as illustrated in Figure 3-27. Extrapolation of the data shows that if the flame temperature could be maintained below 3000° F and the residence time below 10 milliseconds, the NO_x concentration would be less than 25 ppm assuming homogeneous fuel-air mixture burning in the combustor. The data exclude "prompt" NO or fuel bound NO effects

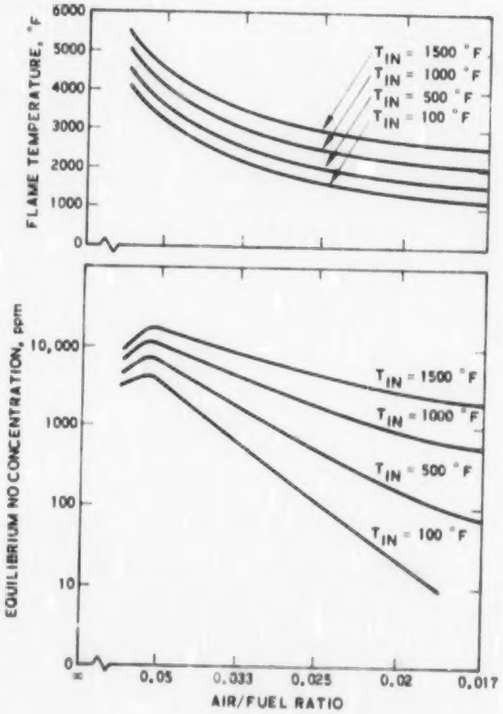


Figure 4-67. Flame temperatures and equilibrium NO concentrations as functions of air-fuel ratio for various inlet temperatures (Ref. 4-106)

which are not greatly temperature-dependent. Nevertheless, the indication is clear that primary combustors operating at fuel to air equivalence ratios of 0.5 or less, homogeneous fuel-air mixtures and short fuel combustion times will form the base for the first two criteria leading to a low emission combustor. In order to approach this goal, liquid fuel must be well atomized for quick vaporization and mixing with the air before ignition.

This leads to the next two criteria for low emission combustion, i.e., fine atomization and premixing. With gaseous fuels there is, of course, no need for atomization but there is still the need for intimate mixing with air before combustion. Compliance with the criteria indicated above will lead, for instance, to the design of a compact, prevaporized, premixed, well-stirred combustor as discussed in various references (for instances, 4-107) for automotive, aircraft, and stationary gas turbines.

A design and a successful development of such a combustor is a complex task. In addition to the consideration of emissions, there is also the overall cycle efficiency problem related to the additional pressure drop in the combustor required for air blast atomization, mixing and recirculation. Also, there is a problem of prevention of combustion flameouts by operating the primary zone within certain fuel-air ratio limits as shown, for instance, in Figure 4-68. Furthermore, in order to prevent preignition in the premix chamber the mixture residence time in the premix chamber must be longer than the evaporation time of the fuel droplets but shorter than the mixture ignition delay (Ref. 4-106). Since the evaporation time of small droplets is typically of the order of 5 ms, and the fuel-air mixture ignition delay is about 20 ms (at 1500° F), the residence time in the premix chamber must be balanced between the two values leaving as great a time margin as possible to prevent preignition (Ref. 4-106). All this points to a

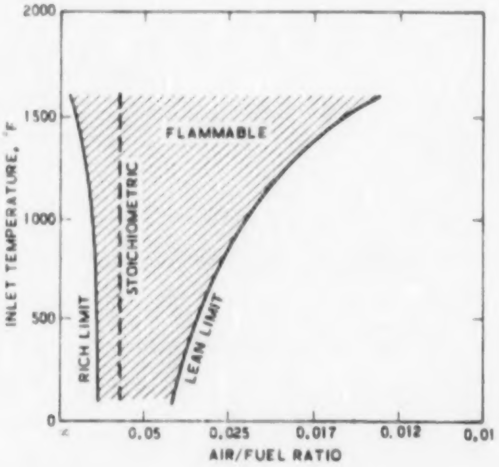


Figure 4-68. Limits of flammability of a paraffin hydrocarbon (C_nH_{2n+2}) showing the influence of inlet temperature (Ref. 4-106)

very careful flow balance which, for automotive gas turbines operating at a variety of loads, may not be possible without some variable control of primary or secondary air flow, and which for stationary units designed for long life will require extensive development and testing.

To comply in the interim with the emission limitations, the stationary gas turbine manufacturers have proceeded in two directions: (1) introduction of modest modifications in the combustor to reduce the emissions as much as possible without prolonged development work, and (2) water or steam injection into the primary zone of the combustor. The effect of water injection will be dealt with in Section 4. 3. 4 while the effects of interim combustor modifications and the farm term development plans are discussed below.

4. 3. 3. 2 Interim Combustor Modification

Of the modest combustor modifications aimed at emission reduction, the following few can be quoted as more significant than others: air-blast or air assist atomization; leaning of the primary zone; early flame quench; and flow recirculation (reverse flow in the primary zone). Application of these modifications to various combustors produced different results due probably to different flow patterns, different primary combustor designs and different degrees of atomization. Air-blast atomization has been shown to be helpful in smoke reduction, and this effect was generally confirmed by various stationary gas turbine manufacturers and users (Ref. 4-108). The effect of the air-blast or air assist approach is related to improving the quality of liquid fuel atomization by reducing the mean Sauter diameter from about 100μ for pressure injection to less than 40μ for air atomization (Ref. 4-107). This results in a reduction of the evaporation and combustion time, and leans the locally rich hot fuel pockets which are smoke and carbon producers. On the other hand, Ref. 4-108 stated that changes that decreased NO_x increased the CO emission and vice versa.

Ref. 4-109 reports test results with reduced flame residence time accomplished by moving the upstream air dilution holes, in order to direct more air and partially burned recirculation gas into the primary zone. The effects of these changes are illustrated in Figure 4-69, showing NO_x for the standard combustor, with 25-percent reduction in flame residence time (Curve II), with primary zone leaned to equivalence ratios of about 0.5 (Curve III) and 0.3 (Curve IV), as a function of the temperature rise in the combustor. It can be seen that the NO_x reduction is more pronounced in the high temperature rise regime (above 1000°F). Similar tests performed in scaled combustors operated at various combustor exit temperatures (operational temperature is 1800°F) show more moderate reductions in NO_x. According to Figure 4-70, showing laboratory model data, progressively leaner primary zones (Mod 1, Mod 2) resulted in NO_x reductions of the order of 10 to 20 percent. Similar reductions were obtained with fuel oil. In these tests some reduction in CO and HC was observed with the leaner primary zone designs. The effect of primary zone gas recirculation is shown in Figure 4-71 for natural gas, showing that the NO_x reduction becomes more pronounced at higher temperatures.

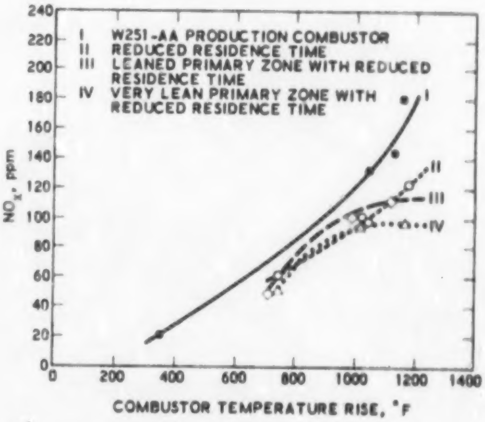


Figure 4-69. NO_x variation with temperature rise for production and modified combustors (Ref. 4-109)

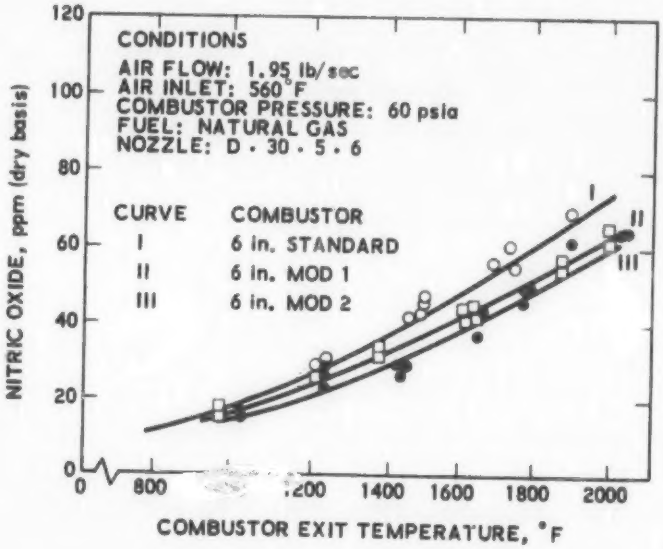


Figure 4-70. Effect of primary zone leaning on NO emission — natural gas (Ref. 4-109)

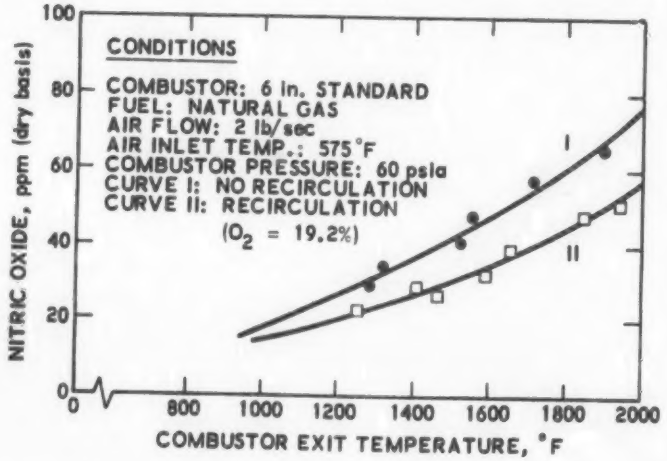


Figure 4-71. Effect of cooled exhaust gas recirculation on NO emission — natural gas (Ref. 4-109)

Again, similar trends were observed with fuel oil. The effect of recirculation on CO and HC was negligible (Ref. 4-109).

Thus, summarizing the possibilities of interim combustor modifications, it appears that the modest modifications of standard combustors, while effective in reducing NO_x (while slightly increasing or not affecting the CO emissions) will not be sufficient to meet any of the regulations quoted in Section 4.3.3.1 at base or near base load conditions. Figure 4-66 indicated that NO_x reduction of over 70 percent is necessary for the average state-of-the-art gas turbine to meet Rule 68 or the proposed EPA limit. However, approximately half of the required improvement might be achieved by modification of the current combustors (air atomization, lean primary zone and gas recirculation). This applies both to natural gas and fuel oil since tighter limits for NO_x are applied to gas.

Of course, emission reduction is desirable, and incorporation of these combustor modifications would permit, for instance, a lower rate of water injection to bring the NO_x emissions within the specified limits.

4.3.3.3 Low Emission Advanced Combustors

Most of the work on advanced gas turbine combustors is being performed on aircraft and automotive gas turbines, and the previous discussion on the design approaches to advanced low emission combustors was based on work on these two types of gas turbines. The stationary gas turbines will directly benefit from this work, but it should not be confined to it since the restrictions on weight and size always associated with mobile applications of gas turbines do not apply to stationary power plants. Most of the domestic stationary gas turbines descend from their aircraft predecessors and thus inherited their in-line compactness which, while indispensable in an aircraft or automobile, is of secondary importance in a stationary power plant. It is quite possible, for instance, that an externally (i.e., not in-line between the compressor and turbine) mounted combustor could possess more flexibility in the incorporation of emission reducing features than a combustor squeezed between the compressor and turbine whose length is kept as short as possible to preserve the required stiffness of the shaft and housing. Thus, while the results presented here are obtained either for aircraft or automotive gas turbines, it should be kept in mind that similar or better results are being obtained, although they are not yet published, on externally mounted combustors. Such combustors are being built by Brown-Boveri for use in Turbodyne Corporation power plants and, reportedly, such plants will meet Rule 68 standards with "dry" (no water injection) operation.

4.3.3.3.1 Solar Combustor

One of the AAPS advanced low emission combustors is being developed by Solar in the form of its jet induced circulation (JIC) combustor (Ref. 4-102 and Ref. 4-107). The schematic of the primary zone of this combustor is shown in Figure 4-72. It contains all the elements of an ideal well-stirred reactor. Air blast atomization of liquid fuel assures small droplet sizes which are vaporized and mixed with air in a premix chamber. The fuel-air mixture emerging from the premix chamber acts as an ejector inducing a high degree of recirculation required for stability at lean equivalence ratios (<0.35 in the primary combustor.) Through this entrainment of hot products, the fuel-air mixture in the jet is effectively ignited. In its alternate design Solar utilizes eight primary air and fuel jets which are inclined by 45° in the upstream direction. The jets impinge in the center of the combustor thus providing a motive jet for the recirculation flow.

The emissions from this combustor, operated with kerosene fuel, were typically less than 1.0 lb NO_x/1000 lb fuel (<10 ppm), less than 0.5 lb CO/1000 lb fuel (<10 ppm) and less than

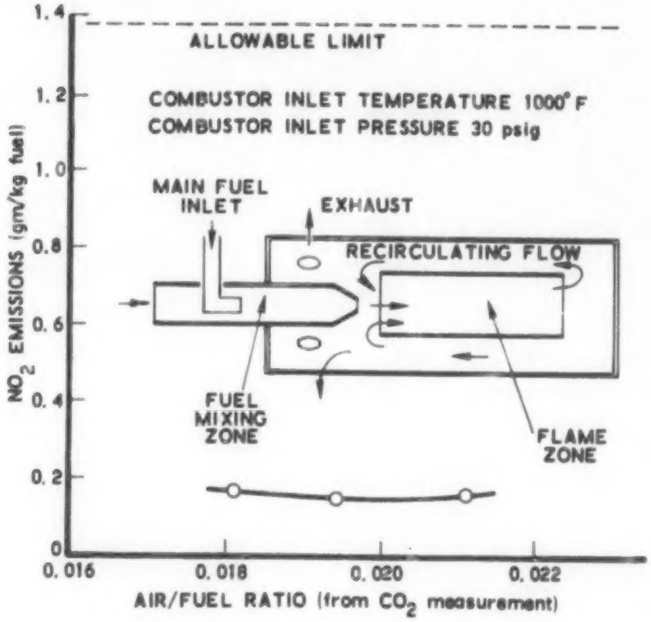


Figure 4-72. High recirculation stabilized lean primary zone combustor schematic and NO_2 emission (Ref. 4-107)

0.5 lb HC/1000 lb fuel (<20 ppm). If such a combustor could be developed for stationary gas turbines, it would meet all of the current regulations with "dry" operation.

4.3.3.3.2 GM Combustor

Another advanced combustor (External Recirculation Combustor) under development at GM for automotive engines is described in Ref. 4-110 and is shown in Figure 4-73. This combustor incorporates all the criteria discussed before: lean primary zone with flame temperature not exceeding 2400°F; short residence time; fuel vaporization and premixing upstream of the combustor. The flame stability and further emission control are enhanced by hot gas recirculation.

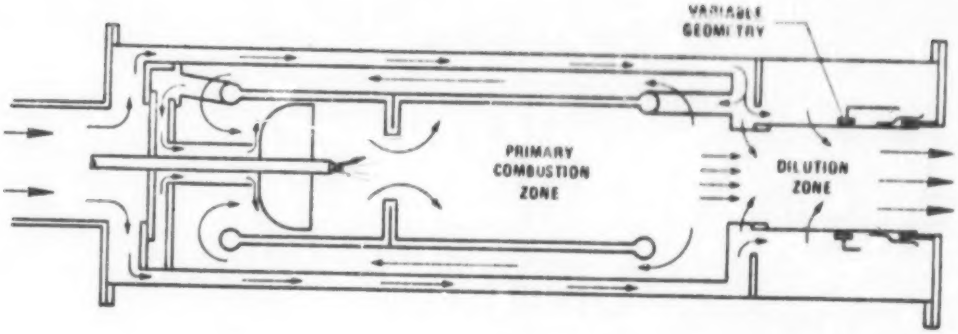
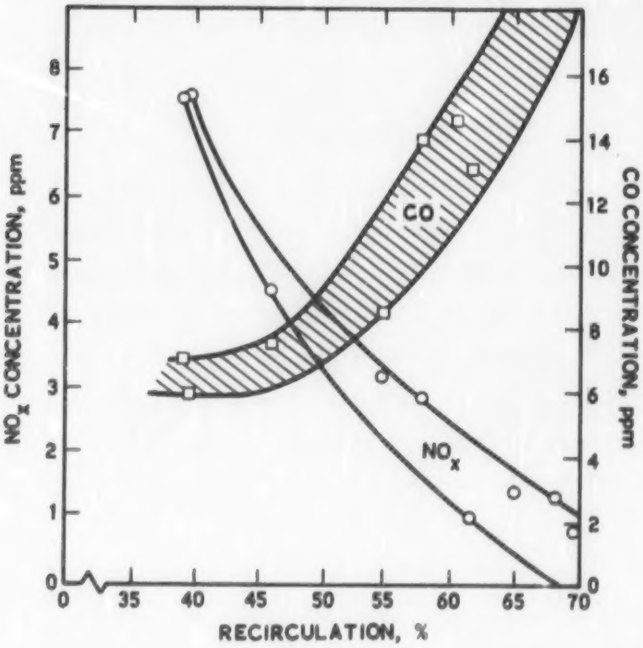


Figure 4-73. External recirculation combustor (Ref. 4-110)

The effects of this recirculation on the NO_x and CO emissions are shown in Figure 4-74. The level of NO_x is drastically reduced and while the emission of CO increases with increasing recirculation, the maximum level is still very low (less than 20 ppm). Above one-half load, the NO_x level is less than 2 lb/1000 lb fuel (<20 ppm), CO is less than 4 lb/1000 lb fuel (<70 ppm), and HC is less than 0.5 lb/1000 lb fuel. It should be noted that this combustor, as well as Solar's combustor, require air flow controls (variable geometry) to meet the varying load demand of automotive engines.

The emissions of the GM-DDA (Detroit Diesel Allison) advanced combustor and their comparison with today's conventional gas turbines was illustrated in Figure 3-36. Of course, the advanced combustor data are only rig test results and there is a long term development work ahead before they become characteristic of an operational gas turbine.

Extensive investigation of low emission gas turbine combustors was undertaken by DDA for an U.S. Army application in a light duty helicopter (Ref. 4-111). The work consisted of an interim modification of the combustor along the lines discussed in Section 4.3.3.2 and of a preliminary evaluation of certain design features



NOTES:

CYCLE POINT 3 (1308°F bit, 22 psia)
PZ TEMP. ≈1820 - 1840°F
F/A OVERALL ≈0.0058

HYDROCARBON LEVELS WERE LESS
THAN 0.4 ppm AT ALL RECIRCULATIONS

Figure 4-74. Effects of recirculation on emissions (Ref. 4-110)

for an advanced combustor. The interim combustor modifications resulted in approximately 50 percent emission reduction. The long range design changes included introduction of a pre-chamber in which the fuel was prevaporized and premixed. This was similar to the approach taken by the other investigators discussed in this section.

4.3.3.3.3 Ford Combustor

Comprehensive analyses of advanced low emission combustors were performed by the Ford Motor Company and test data confirmed the soundness of the analytical approach (Refs. 4-106 and 4-112). Figure 4-75 shows a schematic of Ford's advanced low emission combustor (Externally Vaporizing Combustor, EVC) which incorporated all the previously discussed features. The steady state CO and NO_x emissions of this combustor were less than 1 lb/1000 lb fuel at temperatures of 3000°F (Ref. 4-112). The level of HC was insignificant.

4.3.3.3.4 Aerojet Combustor

Another approach to achieve quick fuel vaporization and fuel-air mixing is the Aerojet Liquid Rocket Company platelet gas turbine (PGT) combustor (Ref. 4-102). In this concept, fuel is uniformly injected from many small orifices into the throats of atomizing venturis through which the primary zone air is passed. The resulting finely atomized spray is rapidly vaporized and mixed with the primary air before combustion at low equivalence ratios with subsequent rapid quenching of the flame by the secondary air. The combustor emissions

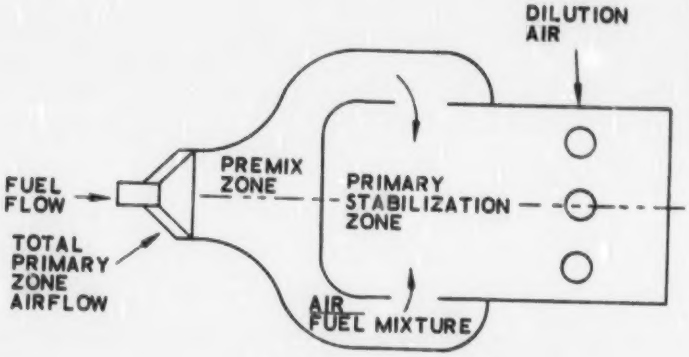


Figure 4-75. Schematic of a low-emission combustor concept — Ford externally vaporizing combustor (EVC) (Ref. 4-106)

achieved to date show NO_x from 0.5 to 2 pounds/1000 pounds fuel and CO from 4.5 to 1.7 pounds/1000 pounds fuel, for primary zone fuel-air ratios of 0.035 to 0.045, respectively.

4.3.3.3.5 GE Combustor

A different type of advanced low emission concept is the surface combustor consisting of either porous plate or catalytic surfaces (Ref. 4-103).

The porous plate combustor is under development by GE. As previously noted, this combustor is aimed at operation below the adiabatic flame temperature. A schematic of a porous plate combustor is shown in Figure 4-76. In this design, the fuel is gaseous or vaporized before being percolated through the porous layer. Flame flashback was one of the problems occurring in this combustor. This problem was alleviated by lowering the equivalence ratio from 0.9 to 0.7 and employing more effective cooling of the porous bed. The measured NO_x emissions were between 0.1 to 1 pound/1000 pounds fuel, and CO between 10 to 40 pounds/1000 pounds fuel, the larger values corresponding to surface air velocities above 20 m/sec. This combustor is applicable only in conjunction with premixed and prevaporized fuels.

4.3.3.3.6 EPA/NASA Combustor

Catalytic surface combustors currently under development by EPA/NASA operate at very lean fuel-air ratios and low flame temperatures as determined by the catalyst bed material. For instance, γ-alumina is limited to 1750° F (Ref. 4-103) while some of the new ceramic materials of monolithic structure have a temperature capability up to 2400° F. The ideal range of operation of catalytic combustors is between 2000 to 2700° F since a rapid decrease in combustion efficiency occurs at catalyst temperatures below 2000° F (Ref. 4-108). The emission of "hot gas" NO_x at the low temperatures ($<2500°$ F) will be

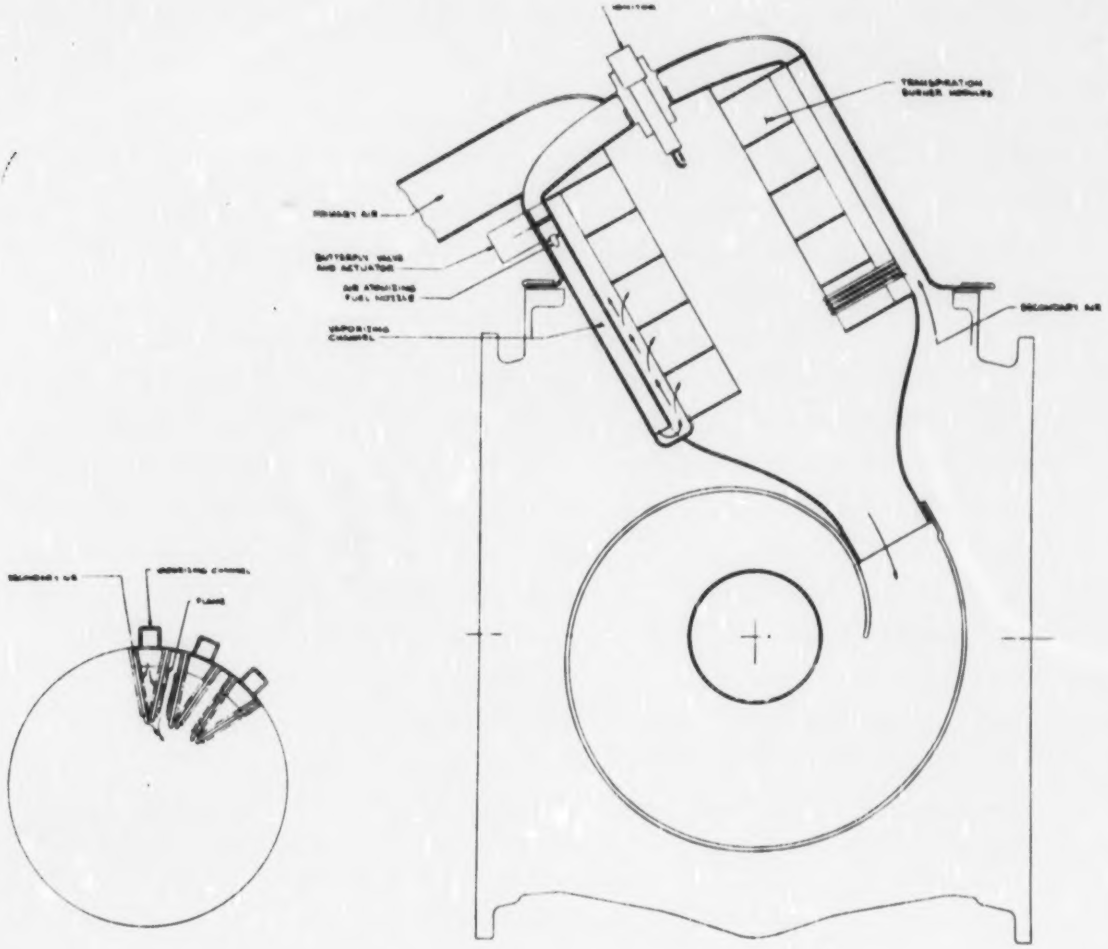

IGNITOR
TRANSPIRATION BURNER MODULES
PRIMARY AIR
BUTTERFLY VALVE AND ACTUATOR
AIR ATOMIZING FUEL NOZZLE
SECONDARY AIR
VAPORIZING CHANNEL
SECONDARY AIR
VAPORIZING CHANNEL
FLAME

Figure 4-76. Transpiration combustor (Ref. 4-102)

significant, but the formation of "prompt" and fuel-bound NO_x in catalytic combustors is open to investigation. Formation of CO and HC is likely to occur under low temperature conditions.

The development work on surface combustion is in the early phase but catalyst durability, long life efficiency and mechanical integrity are emerging as main problem areas.

4.3.4 Water/Steam Injection

Water (or steam) injection in stationary gas turbines is primarily used as a means of NO_x reduction to permit meeting either the Federal or local regulations. Its effect on CO, HC, and smoke varies depending on the particular combustor configuration and the mode of injection. Water injection detracts from the siting flexibility of both simple cycle and regenerative cycle gas turbines in that supply of water has to be assured in quantities about equal to the liquid fuel supply. Furthermore, to avoid detrimental effects on the turbine durability, the water has to be purified to a maximum of 5 ppm (preferably 1 to 2 ppm) of dissolved solids. This additional cost of installation of the water injection equipment, the water treatment plant, and the water itself has to be considered in the economics of stationary gas turbines.

For these reasons, water injection has to be regarded as an interim step in stationary gas turbine emission control until such time when advanced "dry" combustors are fully developed.

The kinetics of water injection on NO_x formation is treated in Ref. 4-113. The main effect of water is the reduction in the flame temperature by the heat required to heat and vaporize the water (which amounts to approximately 30 Kcal/mole). The dissociation energy ($H_2O \rightarrow H + OH$) constitutes only approximately 2 percent of the heat absorbed (Ref. 4-113).

The chemical effect of water injection is minimal. Early in the combustion stage there is a temporary excess of O, OH and H radicals above their equilibrium concentrations and the small amount of dissociated water ($<1/2\%$) has no appreciable effect on this trend. When equilibrium conditions are approached, the water dissociation involving third bodies increases slightly the concentration of O atoms with a compensating effect of a small temperature decrease.

As previously stated, the "hot gas" NO_x formation is controlled by the Zeldovich reactions, particularly the reaction $N_2 + O \rightarrow NO + N$, which is highly temperature-dependent. For instance, reduction of the flame temperature by 200°F reduces the rate of NO_x formation by a factor of three (Ref. 4-114). From the analytical work performed in Ref. 4-113, a flame temperature drop of approximately 350°F was calculated for a water-fuel flow ratio of unity and a combustion zone equivalence ratio of 0.8. Thus, theoretically even greater NO_x reduction rates might be possible. In reality, the reduction is less than theoretical because the water which is atomized into droplets of varying size is not uniformly vaporized and mixed with the gas. The rate of water droplet vaporization is important. Too slow vaporization may produce peak temperatures similar to dry temperatures which would enhance NO_x formation, while very fast vaporization might cause an initial reduction of the temperature which would enhance the formation of CO and HC. The baseline case considered in Ref. 4-113 was with 60μ diameter water droplets which were vaporized in 6×10^{-5} seconds.

Another reason for lower than theoretically predicted reduction in NO_x with water injection is in the fuel bound nitrogen and "prompt" NO_x which may be less temperature dependent than the "hot gas" NO_x. Thus, one would expect that natural gas, which is almost nitrogen-free, should show greater NO_x reduction with water injection than fuel oil which, in turn, would look better than heavy oils and residuals.

Since the heat of vaporization is about one-third of the total heat absorbed by water, steam injection requires greater flow rates than water to achieve the same reduction in flame temperature but the better mixing achieved with steam tends to reduce the difference in the flow rates. This will be shown from some of the industry data. The amount of heat expended on water vaporization is not recovered in the cycle, hence reducing the useful heat input into the turbine. Consequently, a drop in thermal efficiency is expected with water injection while steam injection, which increases the mass flow of the gas turbine without additional compressor work, should improve the thermal efficiency (assuming the steam is provided "free").

Ref. 4-115 quotes a drop in the efficiency of approximately one percent for a water injection rate of one percent of the air flow (one percent of the air mass flow rate is equivalent to about 60 percent of the fuel flow rate), but the power available increases by 3.5 percent for every one percent of water because of the higher mass flow. Ref. 4-116 quotes a larger loss in efficiency with water injection, three percent per one percent of water, and a gain in efficiency with steam of similar magnitude.

As stated before, the effect of water injection depends greatly on the manner in which it is atomized and injected into a combustor. The effectiveness of water injection can be assessed by a water effectiveness factor suggested in Ref. 4-117.

$$\text{water effectiveness} = \frac{(\text{NO wet/NO dry) calculated}}{(\text{NO wet/NOdry) measured}}$$

It was determined in Ref. 4-117 that the water effectiveness factor was 67 percent for gas fuel and 51 percent for fuel oil, probably due to bound nitrogen in the latter as well as better mixing in the former. As an example, Figure 4-77 shows a system where the water is injected shower-like from a toroidal ring around the fuel nozzle. Figures 4-78 and 4-79 show the reduction in NO (expressed as NO_x) for a gas turbine with 1 percent water injection burning fuel oil and natural gas, respectively. The upper curves indicate the NO_x emission with an original unmodified combustor, which was subsequently modified by leaning out the primary zone and by reducing the residence time. This was insufficient to meet the Rule 68 NO_x limits. However, by adding water injection NO_x was reduced by 38 percent for fuel oil and by

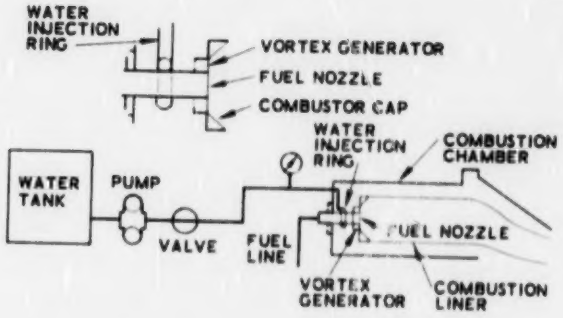

WATER INJECTION RING
VORTEX GENERATOR
FUEL NOZZLE
COMBUSTOR CAP
WATER TANK
PUMP
VALVE
FUEL LINE
WATER INJECTION RING
COMBUSTION CHAMBER
FUEL NOZZLE
VORTEX GENERATOR
COMBUSTION LINER

Figure 4-77. Schematic of water-injection system (Ref. 4-117)

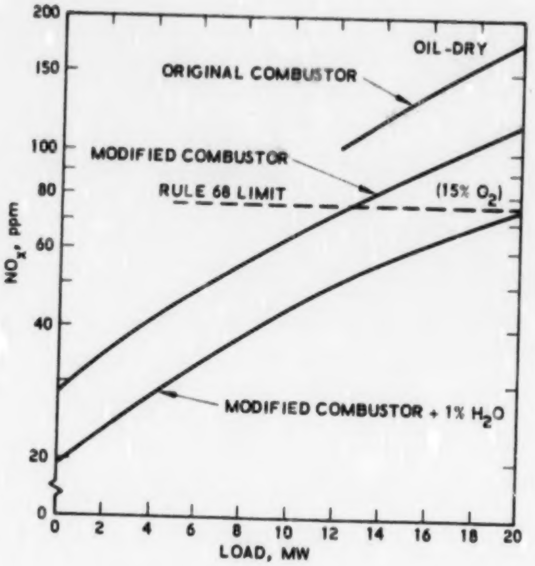


Figure 4-78. Effect of water injection on NO_x emissions from MS 5001 turbine – liquid fuel (Ref. 4-117)

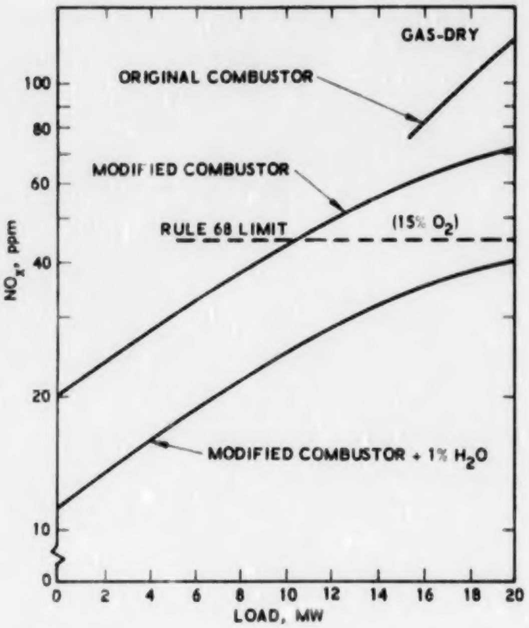


Figure 4-79. NO_x emissions from modified combustion system in MS 5001 gas turbine — gas fuel (Ref. 4-117)

45 percent for gas fuel and the limits were met. CO, HC, and smoke emissions were essentially unaffected by water injection and remained at a very low level.

Ref. 4-118 reported results with various water injection schemes. Injecting water upstream of the fuel injector was successful in reducing NO_x but CO increased by about 200 percent. Injection of fuel and water in a concentric injector was successful not only in reducing NO_x, with CO being essentially unaffected, but also in requiring only approximately one-third of the water flow rate of previous schemes. The emission data with this system are shown in Figure 4-80. As indicated, at a water-fuel ratio of 0.8, the NO_x emissions were reduced by approximately 80 percent.

Figures 4-81 and 4-82 show NO_x reductions achieved with water and steam for oil and gas fired gas turbines. In the case of

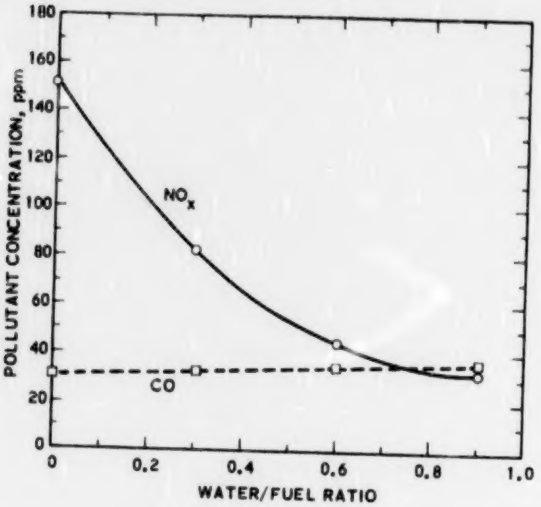


Figure 4-80. Effect of water injection on emissions – 5001K gas turbine engine combustor with fuel oil (Ref. 4-118)

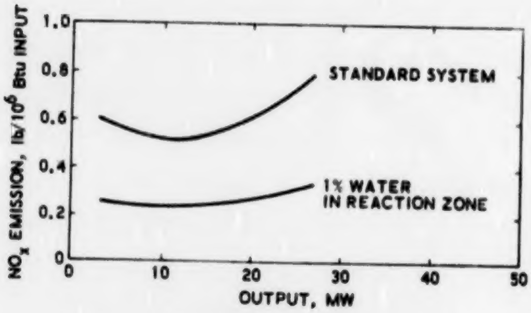


Figure 4-81. NO$_x$ reduction by water-injection, oil-fired Model 5000 Engine, iso-conditions (Ref. 4-116)

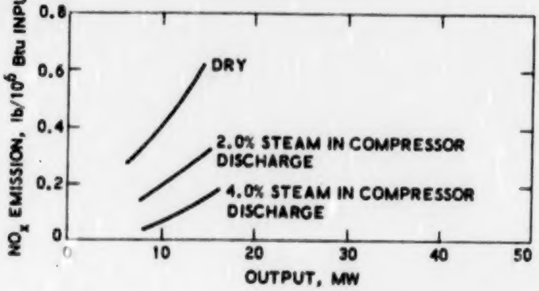


Figure 4-82. NO_x reduction by steam-injection, gas-fired MS-5001L engine, site conditions (Ref. 4-116)

fuel oil a 50 percent reduction of NO_x was obtained with one percent water while the same reduction required about two percent steam. According to Ref. 4-109, six percent steam is required to achieve 55 percent NO_x reduction while 2.5 percent steam resulted in a NO_x reduction of about 20 percent. In this case, 12 percent steam increased CO by approximately 200 percent, with a similar increase in HC.

Different results were obtained in Ref. 4-119 where water injection reduced all emissions. The effect on NO_x is shown in Figure 4-83. Figure 3-34 shown previously, illustrates the reduction of CO; Figure 4-84 shows the effect of HC, and Figure 3-40 shows the smoke reduction. These latter data illustrate the effectiveness of water injection in a combustor in which good matching with water injection rate and mode was obtained.

Another way of introducing water is emulsifying it in liquid fuel, but reportedly low water effectiveness was observed with that scheme.

The use of water injection is now applied in many utilities since it is at present the only known means to enable the state-of-the-art gas turbine combustors to meet the various NO_x standards.

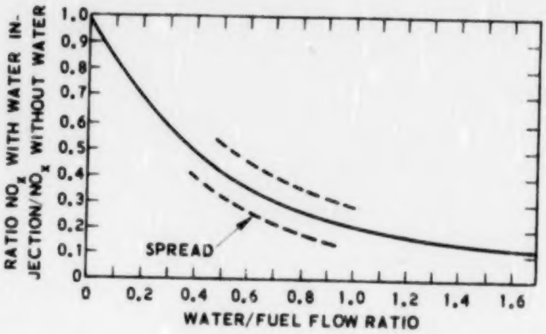


Figure 4-83. Combustion laboratory NO$_x$ reduction with water injection (Ref. 4-119)

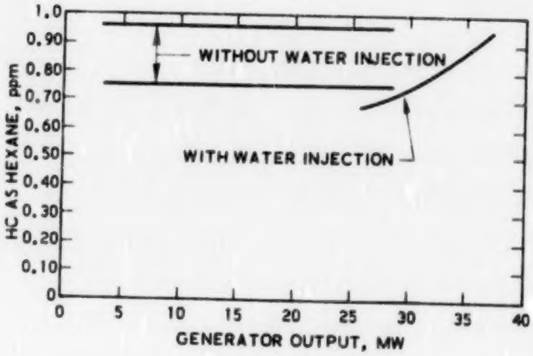


Figure 4-84. Total HC versus load, W-251 engine (Ref. 4-119)

Up-to-date experience does not indicate any deleterious effect of water on turbine life provided the dissolved solids are kept below 5 ppm. If steam injection is used, care must be taken to maintain it in a superheated condition up to the injection point to prevent condensation and slugs of water entering the combustor. At least one case of turbine bucket damage resulting from water condensation has been reported.

In summary, in the case of fuel oil, water injection with rates less than or equal to the fuel flow rate can reduce the NO$_x$ emissions by 35 to 50 percent in low emission combustors and 50 to 75 percent in a standard combustor. Greater reduction (60 to 90 percent) can be expected with natural gas (Ref. 4-113). The level of CO and HC can either be reduced, remain the same, or increase depending on the way the water is injected. If steam injection is used, higher (by a factor of 2) flow rates than water are required and an improvement in the thermal efficiency of the cycle can be expected. A summary of the effectiveness of various emission control techniques is presented in Table 4-12.

4.3.5 SO_x Emission Control

It was stated before that sulfur oxides are formed from the sulfur in the fuel. To comply with the proposed regulations the amount of fuel sulfur for liquid fuels should be <0.8 percent by weight. Estimates by fuel specialists indicate that this limit can be met by all of the available distillates produced in this country, by 75 percent of

TABLE 4-12. EFFECTIVENESS OF VARIOUS GAS TURBINE EMISSION CONTROLS

Design Modifications	Emissions			
	NO_x	CO	HC	Smoke
Interim				
Primary Zone Leaning	10 - 30% reduction	Small reduction	Small reduction	Reduction
Combustor Gas Recirculation	~30% reduction	Negligible effect	Negligible effect	Negligible effect
Water/Steam Injection	50 - 75% reduction (oil) 60 - 90% reduction (gas)	Some reduction or increase	Small reduction or small increase	Small increase possible
Advanced Combustion				
Premixed, Prevaporized, Well-Stirred, External Combustors	20 ppm achievable	40 ppm achievable	5 ppm achievable	Invisible

the crudes, 50 percent of the blends, and about 15 percent of the residuals (Ref. 4-120). If more stringent limits for SO_x are contemplated, or if more of the crudes, blends and residuals are to be used for stationary gas turbines, then two ways are open for SO_x reduction: one is improvements in the refinery process which would reduce the fuel sulfur to an acceptable level; and the other is flue gas treatment to remove the SO_x. The refinery technology is available to achieve sulfur reduction (Ref. 4-121) but at an increase in fuel costs. Various flue gas treatment methods are now in development for steam boilers which will substantially increase the investment and operating costs of the plant. Application of these techniques to stationary gas turbines would be even more costly because of the higher gas turbine flow rates and exhaust temperatures relative to steam power plants. A brief review of the flue gas treatment method is given below since in principle it is applicable to stationary gas turbine exhaust.

In principle there are two different ways of SO_x removal. In one approach, SO_2 is recovered to form a useful product (such as sulfuric acid), and in another, a solid waste is formed. Both, the recovery and throwaway methods can be carried out in either wet or dry systems (Ref. 4-122).

There are more than 50 individual processes known for SO_x removal but only eight have found some degree of acceptability by utility companies. Most of these processes are wet processes consisting of wet lime, alkali or magnesium oxide scrubbing. One process consists of acidified water scrubbing and a dry catalytic oxidation system (Ref. 4-122). Most of the processes produce solid waste which poses a disposal problem. An exception to this is the magnesia process in which the products, magnesium sulfite and sulfate, are used for regeneration of the magnesium and formation of sulfuric acid. The acidified water process produces gypsum, and the catalytic oxidation procedure produces sulfuric acid. Currently the reliability of the various processes is low (maximum 33 percent), mainly because of corrosion, scaling, and plugging problems (Ref. 4-122).

The investment cost of SO_x scrubbing equipment is \$80/kw for a 550 MW steam unit and for future larger units the cost is estimated at \$50/kw which adds about 25 percent to the investment cost of a steam plant. The operating costs are estimated at 0.7 mills/kwh including sludge disposal with additional three percent of energy consumption (Ref. 4-123). As noted previously, the cost figures for gas turbine installations would be higher than for steam power plants.

Summarizing, it appears that for liquid fuels it would be more economical to remove any excess sulfur during the refining process. If this is not possible (for coal, for instance) the flue gas treatment would have to be employed. However, for steam plants this process will not be developed until the 1975 to 1977 time period (Ref. 4-120) and more time would be required for gas turbines. In any case, SO_x control causes an increase in the investment and/or operating cost of the plant.

4.3.6 Smoke, Particulates, and Odor Control

As discussed in Section 3.3.3.3 smoke (which consists of small carbon particles), particulates (consisting of carbonaceous materials as well as burnt fuel particles), and odor formation depend largely on the combustion efficiency. An efficient combustor, which is characterized by low CO and low HC, will also have inherently low smoke and particulates, and in most cases, low odor (also a function of fuel aromatics). Thus, the development of low emission combustors, discussed in Section 4.3.3, will not only maintain low levels of NO_x and CO but will reduce the other emission species as well, with the exception of the fuel-dependent SO_x.

The interim, near-term combustor modifications, as discussed in Section 4.3.3.2, were, in general, successful in reducing smoke and particulates to an acceptable level (Ref. 4-124). Consequently, the use of manganese in the form of methylcyclopentadienyl manganese, which was found very successful in smoke reduction (Refs. 4-124 and 4-125), is usually not required.

There are no reports on the effect of the interim modifications on odor which, in itself, is not a problem in stationary gas turbines, but judging from Ref. 4-126, any reduction in HC should have a beneficial effect on the odor.

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SECTION 5

EMISSION CONTROL SYSTEM ASSESSMENT

This section of the report presents an assessment of the emission control techniques/systems identified in Section 4 relative to their applicability to new and existing stationary engine installations. In addition, preliminary information on the economics of emission control in stationary engines is included in this section.

Subsection 5.1 is concerned with diesel engines, and Subsections 5.2 and 5.3 are devoted to spark ignition engines and gas turbines, respectively.

5.1 DIESEL ENGINES

The following discussion is concerned with an evaluation of techniques/devices potentially applicable to diesel engines. Since NO_x is the predominant exhaust pollutant from these engines, the discussion will be concerned primarily with that particular species. The most promising NO_x abatement techniques are identified and system cost data are presented whenever possible.

5.1.1 Emission Control Techniques/Devices

5.1.1.1 Engine Derating

In general, reducing the engine load at rated speed increases the specific mass emissions of NO_x and HC of naturally aspirated, open-chamber and turbocharged divided-chamber diesel engines. Conversely, slight reductions in NO_x have been observed in the case of turbocharged, open-chamber diesels. In some engines, simultaneous reduction of engine speed and load might result in substantially lower NO_x, CO, and smoke emission levels.

Although insufficient information is currently available to permit a quantitative evaluation of this approach, it appears that limited engine derating might be practical for some engines, particularly in conjunction with other emission control techniques. Basically, engine derating could be implemented in both new and existing engines, but a change in gears would be generally required to compensate for the lower engine speed. However, engine derating raises the engine investment cost per horsepower and has some impact on the specific fuel consumption, maintenance requirements, and life of the engine. Obviously, all these parameters would have to be considered in a meaningful cost effectiveness evaluation of this technique, relative to other emission control approaches.

5.1.1.2 Intake Manifold Temperature

Reduction of the air intake temperature has a beneficial effect on the NO_x emissions from diesel engines. Although the effect is rather small (20 percent reduction per 100°F temperature drop), it is accompanied by a slight improvement in specific fuel consumption and essentially no change in the HC, CO and smoke emissions.

Intake charge cooling, which would be particularly effective at high ambient temperature conditions requires the installation of a heat exchanger upstream of the engine. In turbocharged diesels, intercooling is a proven technique to increase the power output capability of the engine and to lower its specific fuel consumption. Additional improvements in engine performance and NO_x might be accomplished by increasing the size and/or effectiveness of the cooler and by employing air or city water as coolants. This approach merits further investigation, particularly for stationary engines where the volume constraints of the air cooler would be less severe than in mobile installations and where city water is available. It is applicable to both new and existing engines and might be particularly attractive in conjunction with other emission control techniques, such as injection timing retard or EGR.

5.1.1.3 Fuel Injection Timing Retard

The effect of fuel injection timing retard on diesel emissions has been determined experimentally by many investigators and the available test data are in reasonable agreement for all diesel engine classes. Particularly the first few degrees retard are very effective in reducing NO_x at the expense of moderate increases in CO, smoke, and specific fuel consumption and a small loss in output power. HC either increases, decreases, or remains constant with increasing timing retard, but the variations are generally less than ±70 percent. For most engines, injection retard is probably limited to about six degrees or less because the specific fuel consumption deteriorates rapidly beyond that point. Typically, six degrees injection retard lowers NO_x by about 40 percent while CO and exhaust smoke increase by about 50 percent or less. Since CO and smoke emissions are generally low in well maintained diesel engines, this increase is probably tolerable in most cases.

If stringent emission standards would be promulgated for stationary diesel engines, it appears that most manufacturers might incorporate some degree of injection retard. This technique requires no hardware changes and can be easily incorporated into new and existing engines. The associated loss in specific fuel consumption might be alleviated to some degree by simultaneously optimizing the timing retard and injection period schedules. Potential problem areas related to injection timing retard include exhaust valve and turbocharger durability which might be adversely affected by the higher exhaust gas temperatures obtained with retarded timing.

5.1.1.4 Combustion Chamber Modifications

Optimization of the combustion chamber geometry in terms of bowl diameter to bore ratio, piston to head clearance, compression ratio, and air swirl can result in lower NO_x emission and specific fuel consumption of diesel engines. However, the combustion cycle is presumably optimized for specific fuel consumption, and any reduction in NO_x would be accompanied by an increase in fuel consumption relative to the optimum condition. Although the observed effects are rather small, optimization of these parameters is desirable and could be fairly easily implemented in new engine designs at no increase in hardware cost. Conversely, such modifications are not considered to be cost effective for retrofit applications.

5.1.1.5 Injection System Modifications

A number of fuel injection system parameters, including injection rate, orifice size, spray angle, and sac volume are important design parameters, impacting the emission characteristics of diesel engines. On one engine, optimization of the fuel injection rate has resulted in a 20-percent reduction in NO_x with no attendent loss in fuel economy. However, incorporation of this approach might require a new fuel injection system which would be capable of withstanding the higher stress levels associated with shorter injection periods. Optimization of the fuel injector spray angle and orifice size is expected to cause moderate reductions in NO_x and HC.

Basically, these techniques are considered to be applicable to new engines and retrofit installations alike. However, a meaningful evaluation of the cost effectiveness of these modifications, as applied to the various engine categories, is not possible at this time because of a lack of applicable test data.

5.1.1.6 Fuel Effects

Variations in the fuel composition and cetane number have some effect on diesel engine emissions. In medium speed naturally aspirated and turbocharged diesels, NO_x, HC, and CO tend to decrease with increasing fuel cetane number. However, it appears that the observed reductions (about 10 percent) are not sufficiently large to warrant the manufacture of higher cetane fuels for use in these engines. Conversely, a reduction in cetane number substantially below current levels could result in markedly higher HC, CO, and NO_x emissions.

Fuel bound nitrogen has been identified to be a significant contributor to the total NO_x emissions from oil- and coal-fired boilers. Although similar effects might be obtained in diesel engines when operating on heavy distillate and residual fuels, there is insufficient data available at this time to permit a meaningful assessment of this phenomena.

Smoke suppressant fuel additives, although quite effective in reducing smoke, are not recommended by most manufacturers because of potential adverse effects on engine durability and human health. Work on odor suppressant additives has not been successful.

5.1.1.7 Fumigation

Based on a very limited data sample it appears that small reductions in NO_x might be achieved in diesel engines by means of fumigation (injection of a small amount of fuel into the intake system). This method could be incorporated into both new and existing engines. However, additional development programs would be required to provide the data needed for a meaningful assessment of this particular technique.

5.1.1.8 Exhaust Gas Recirculation

Exhaust gas recirculation has been shown to be a very effective NO_x abatement technique for all types of diesel engines, particularly in those cases where the EGR flow is cooled before admission into the intake manifold. However, incorporation of EGR generally causes some increase in CO, smoke, and specific fuel consumption. accompanied by moderate reductions in the HC emissions and engine power output capability. At rated engine load incorporation of about ten percent cooled EGR typically results in a 50-percent reduction in NO_x, accompanied by a 100- to 150-percent increase in CO and smoke, a 1.5-percent increase in SFC, and some loss in engine power output. EGR is generally less effective at part load conditions because of the attendant increase of the oxygen concentration in the engine exhaust.

Most manufacturers feel that the application of EGR would create a number of potential problem areas in the engine. These include corrosion and deposit buildup in the EGR circuit, particularly the EGR cooler, as well as excessive engine wear and fuel oil contamination resulting from the sulfur and metallic compounds contained in the fuel. Additional difficulties might arise in turbocharged and aftercooled engines, due to deposit buildup in the compressor and heat exchanger. The deposit-related problems might be alleviated by incorporating a filter system which would be serviced periodically.

In principle, EGR could be added to both new and existing engines. However, considerably more testing would be required to permit a meaningful assessment of its long term effects on engine durability, performance and emission control effectiveness.

5.1.1.9 Water Injection

Although the available test data show considerable scatter, there is no doubt that water injection into the intake system (induction) is an effective method to reduce the NO_x emissions from diesel engines. The principal advantage of water injection relative to many other techniques is the fact that the improvement in NO_x can be accomplished with little or no loss in specific fuel consumption.

Typically, water induction at a rate about equal to the fuel injection rate reduces the NO_x emissions by about 50 percent. Based on a very limited data sample it appears that HC increases slightly with increasing water injection. Conversely, smoke shows a tendency to decrease with increasing water flow rates, whereas CO and SFC show very little change with water flow rate.

Water induction could be added fairly easily to new and existing engines, but a number of potential problem areas would have to be resolved before this technique could be seriously considered for use in stationary engines. These include corrosion and wear of the intake system, intake valves and water injection nozzles, as well as degradation of the lube oil. Although these problems might be alleviated by using distilled or demineralized water, the related cost increase would have to be considered when comparing this approach to other potential control techniques.

Although water injection in the form of emulsified fuel has not been successful in one engine, this technique merits further consideration primarily because the previously noted corrosion and wear problems in the intake system would be eliminated. However, wear and corrosion in the fuel system are considered to be potential problem areas. Furthermore, only a single set of injectors would be required which would be beneficial from a cost point of view.

5.1.1.10 Catalysts

When fresh, catalytic converters are quite effective in the control of HC, CO, and odor from diesel engines. However, the durability of these catalysts has not been demonstrated, particularly in conjunction with low grade fuels. More importantly, there are no known NO_x reduction catalysts that would be effective in the oxidizing atmosphere typical of diesel exhaust. Although theoretically possible, generation of sufficient amounts of reducing species such as CO and H_2 by means of afterburning is not considered to be economically feasible at this time.

Since the HC and CO emissions are generally low in diesel engines, it appears that the use of oxidation catalysts would be limited to those cases where excessive amounts of HC and CO would be generated as a result of the application of NO_x abatement systems.

5.1.1.11 Thermal Reactors

Because of the low HC and CO concentrations in the exhaust of diesel engines and the relatively low exhaust gas temperatures, thermal reactors would probably be rather ineffective in terms of HC and CO abatement. Furthermore, thermal reactors have no effect on NO_x, the principal pollutant species emitted from diesel engines.

5.1.1.12 Turbocharging

In general, incorporation of a turbocharger into naturally aspirated diesel engines increases the specific mass emissions by as much as 70 percent while lowering CO, smoke, and SFC. However, by combining turbocharging with retarded injection timing and intercooling, NO_x reductions up to 35 percent have been demonstrated without occurring any loss in specific fuel consumption relative to equivalent naturally aspirated engines.

Basically, turbocharging/intercooling is applicable to both new and existing diesel engines and is being seriously considered by many manufacturers of automotive diesels to meet future emission control standards. For durability reasons, the compression ratio of existing engines might have to be lowered to compensate for the higher combustion processes obtained with turbocharging.

5.1.2 Emission Control Systems

To date, only a very limited amount of research work has been conducted to characterize and optimize the effectiveness of promising emission control systems for potential use in both new diesel engines and retrofit installations. In general, the emission reductions achieved with these control systems are in reasonable agreement with projections, as determined from the available test data of the individual devices/techniques.

Best results, in terms of NO_x reduction and attendent SFC loss, have been reported for a control system consisting of ten percent EGR and intake air cooling. In this case, NO_x was reduced by about 40 percent while the specific fuel consumption improved by almost two percent. Basically, a control system of this type could be incorporated fairly easily into new and existing engines. However, it should be emphasized again that these results are for new control systems and do not include any allowance to account for potential performance degradation of the system.

According to one diesel engine manufacturer, even more favorable NO_x versus specific fuel consumption tradeoffs might be realized by means of extensive engine modifications and optimization of certain engine components.

5.1.3 Economic Considerations

5.1.3.1 Emission Control Equipment and Maintenance Cost

In view of the very limited emission control system work conducted to date, it is impossible at this time to perform an accurate assessment of the initial cost and maintenance requirements of the various potential emission control devices and systems.

With respect to EGR, fairly reliable cost figures are available for automotive systems which have achieved an advanced state of development. However, for a number of reasons these data are not considered to be applicable for liquid-fueled diesel engines. First, the EGR flow used in diesels would have to be cooled to achieve acceptable effectiveness and this would add substantially to the complexity and cost of the system. In addition, because of the higher corrosiveness of the diesel exhaust gases relative to spark ignition engines, there is considerable uncertainty at this time regarding the degree of EGR filtering required to minimize deposit buildup in the EGR cooler and wear of certain engine parts. Also, incorporation of EGR tends to degrade the output power capability of the engine, thus further raising the engine cost per horsepower output.

One manufacturer has stated that incorporation of a turbocharger increases the cost of a medium size diesel engine by about 10 percent, or \$2.50 to \$3.00 per horsepower. In large stationary diesels (typical cost about 100 \$/hp), the percentage increase is somewhat lower. Addition of an intercooler would contribute another 30 to 50 cent per horsepower to the overall cost of the engine. However, since intercooling improves the specific fuel consumption of the engine and its power output capability the cost of the intercooler would be recovered within a relatively short time.

No initial cost and maintenance figures are currently available for water injection systems. It is estimated that the initial cost of such a system would be comparable to the cost of the fuel injection system. Depending on the required purity of the water, additional cost penalties would be due to the manufacture of the purified or distilled water and the cost of construction of the purification plant. Pertinent water cost data are presented in Section 5.3 for gas turbines.

One engine manufacturer has estimated that incorporation of a typical emission control system would increase the initial cost of the engine by five to ten percent, or \$1.25 to \$3.00 per horsepower. In addition, the maintenance cost of the engine would increase by about 10 to 15 percent. These estimates are corroborated by another manufacturer who indicated that the initial cost of emission control hardware would not greatly affect the sales price of the engine. However, any increase in specific fuel consumption as a result of emission control would have a significant impact on the operating cost of the engine. This becomes apparent when considering the operating schedule of engines employed in continuous power installations. Based on the assumption of a fuel cost of six cents per pound, and an engine operating time of 7000 hours per year, the total fuel cost per horsepower is about $170 per year, or $5100 for 30 years, which represents the average life of large stationary diesel engines. Hence, in this case, each percentage point loss in specific fuel consumption is equivalent to about $50 per horsepower over the life of the engine. Of course, the importance of fuel cost decreases with decreasing operating time of the engine.

Although fuel cost appears to be the controlling parameter for many installations, the effect of the emission control system on engine life, maintenance requirements, and parts replacement would have to be determined before an accurate economic analysis of diesel engine emission control would be possible.

5.1.3.2 Operating Cost

The estimated cost of NO_x control due to changes in engine specific fuel consumption in terms of dollars per ton NO_x removed versus NO_x emission level for turbocharged open-chamber diesel engines is presented in Figure 5-1. Maintenance cost differentials and water cost are not included in the figure.

The curves plotted in Figure 5-1 are based on the SFC vs NO_x reduction data shown in Figure 5-2 and were computed from the following equation, using a fuel cost of $0.06 per pound, a baseline NO_x level ($NO_{x,o}$) of 12 g/bhp-hr and a baseline specific fuel consumption ($BSFC_o$) of 0.380 lb/bhp-hr.

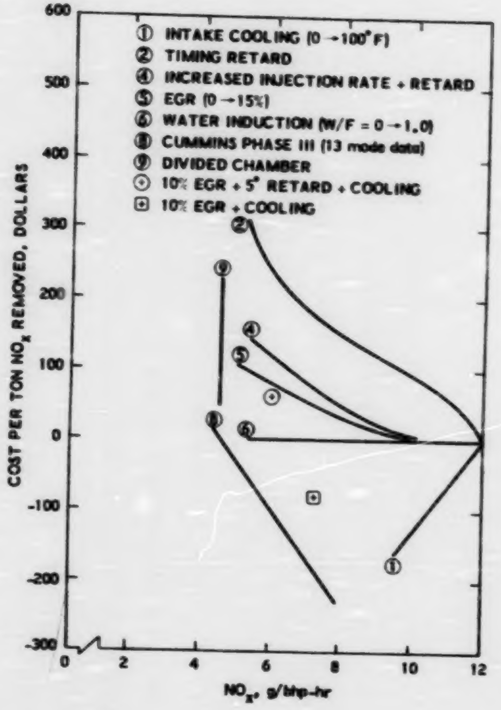

Figure 5-1. Projected NO_x abatement cost - turbocharged, open - chamber diesels (fuel cost only)

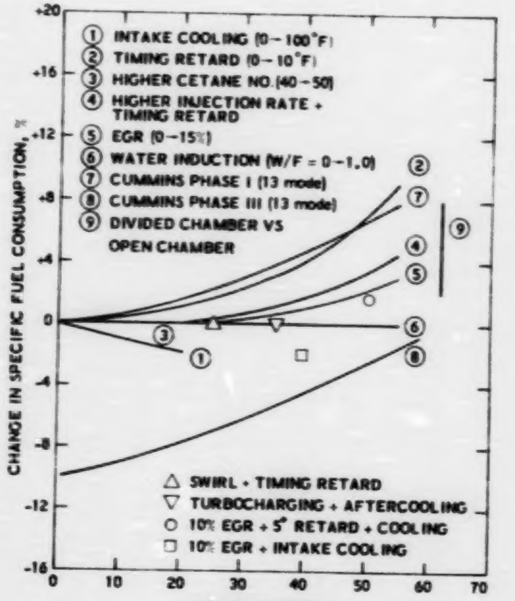

Figure 5-2. Projected average NO_x vs SFC correlations

$$\frac{\$}{\text{Ton } NO_x} = 0.908 \times 10^6 \frac{BSFC_o}{NO_{x,o}} \times \$/\text{LB fuel}$$

$$\times \frac{\left(\frac{BSFC}{BSFC_o} - 1\right)}{\left(1 - \frac{NO_x}{NO_{x,o}}\right)}$$

The curves presented in Figure 5-2 represent averages for the various techniques discussed in Section 4.1.

Based on the above equation, the fuel related NO_x abatement cost is proportional to the specific fuel consumption and inversely proportional to the NO_x emission level of the uncontrolled engine. Hence, the NO_x abatement cost for other diesel engines (naturally aspirated and divided chamber) can be easily determined by ratioing the cost data plotted in Figure 5-1 by the baseline BSFC and NO_x levels of these engines.

As indicated in this figure, timing retard is the least cost effective NO_x abatement technique. Conversely, the most cost effective approaches considered to date are the Phase III modification under development by Cummins and a system consisting of ten percent EGR and intake air cooling.

5.1.4 Promising Emission Control Systems

Based on the evidence presented in the previous subsections and in Section 4.1, the following approaches offer the best compromise between NO_x emission control and specific fuel consumption degradation.

5.1.4.1 Near Term Controls

Of all known emission control techniques for diesel engines, fuel injection timing retard is the only potential NO_x abatement method that involves no hardware changes. However, because of substantial losses in specific fuel consumption, timing retard is also the least cost-effective technique considered to date for use in diesel engines. Some improvement in specific fuel consumption might be achieved by employing optimized timing retard and injection period schedules. This combined approach, which might require new fuel injectors, is considered to be applicable to both new and existing engines.

Intake air cooling, although not extremely effective by itself, appears to be very attractive for use in conjunction with other control techniques, such as timing retard, EGR, or water injection. In this case, the increase in fuel consumption due to timing retard and/or EGR would be compensated for, at least in part, by the reduction in SFC obtained with charge cooling. Manifold cooling of the air by means of a water-to-air or air-to-air heat exchanger would be particularly effective for turbocharged diesels because of the relatively high temperature rise in the compressor. In some large diesel engines in which intercooling has been employed for some time, further NO_x reduction might be achieved by improving the effectiveness of the heat exchanger.

Water induction into the intake manifold of the engine or injection into the cylinder either directly or in the form of emulsified fuel are other potential near-term NO_x abatement techniques for use in new and existing engines. These methods merit further investigation, particularly with regard to the effects on engine life and specific fuel consumption. The NO_x reduction effectiveness of water induction has been demonstrated in limited test work, but consistent data on specific fuel consumption are lacking as are data on emulsified fuels.

5.1.4.2 Far Term Controls

As indicated in Figure 5-1, a new engine design currently being developed by one manufacturer represents the most cost-effective NO_x abatement approach projected for diesel engines to date.

In this design, NO_x control is accomplished by means of limited timing retard combined with many engine component modifications optimized for low emission operation. Further NO_x reduction, possibly at the expense of a slight increase in specific fuel consumption and some of the other pollutants, might be achieved by adding intake air cooling, EGR, or water injection. However, as previously stated, the long term effects of EGR and water injection would have to be determined experimentally, before these particular techniques could be safely incorporated into stationary engines.

Because of the inherently lower NO_x emissions of prechamber diesel engines, relative to open chamber configurations, it appears that those engine design and operating parameters having the greatest impact on the formation of NO_x and the specific fuel consumption of prechamber engines should be optimized. Again, intake air cooling, combined with EGR or water injection, might further improve the cost effectiveness of NO_x abatement and these techniques merit additional research and development.

5.2 SPARK IGNITION ENGINES

5.2.1 Emission Control Techniques/Devices

5.2.1.1 Air-Fuel Ratio

One of the most important engine operating parameters relative to exhaust emission control is the air-fuel ratio of the combustible mixture. Relative to automotive engines, substantially lower HC, CO, and NO_x emissions and lower specific fuel consumption can be achieved in stationary engines by operating these engines with excess air.

Very low HC and CO emissions have been demonstrated in stationary gas fueled spark ignition engines operating with 30 percent to 50 percent excess air. However, the observed NO_x emission is very high in these engines, probably because of high local combustion temperatures resulting during the combustion process in large engines and the relatively long residence time of the reacting species in the high-temperature post-flame zone. In this case, further leaning of the air-fuel mixture might result in lower NO_x at the expense of higher HC and fuel consumption.

Stationary gasoline engines operate generally in the rich or near stoichiometric air-fuel ratio regime. As a result, HC and CO are higher than in gas engines but NO_x is lower. In order to minimize these pollutants, gasoline engines should be operated with at least 30 percent excess air. However, achievement of stable engine operation under these conditions would require uniform air-fuel mixture.

5.2.1.2 Ignition Timing

Spark timing retard from the maximum brake torque setting results in moderate reductions of NO_x and HC, accompanied by a loss in engine power and fuel economy. Furthermore, at high engine loads, overheating and burn-out of exhaust valves may occur in the case of retarded timing. For these reasons, this approach is not considered to be a viable emission control technique for stationary spark ignition engines.

5.2.1.3 Mixture Temperature

Lowering the mixture temperature results in a decrease of the NO_x specific mass emission from spark-ignition engines. In this case, there is no effect on CO but a slight increase in HC might occur in gasoline fueled spark-ignition engines. In general, lowering the mixture temperature has a beneficial effect on fuel economy, volumetric efficiency, and fuel octane requirement of spark ignition engines.

Reduction of the mixture temperature in gasoline engines can be accomplished by eliminating inlet manifold heating and in gas engines by passing the inlet air through an evaporative cooler. In this case, NO_x would be even further reduced because of the increased moisture content of the inlet air.

5.2.1.4 Coolant Temperature

Engine tests have indicated that increasing the engine coolant temperature results in considerably lower HC emissions. However, NO_x might increase slightly, particularly at lean mixture operation. Thus, this technique appears to be feasible only in combination with other control methods.

Conversely, lowering the engine coolant temperature might be feasible to achieve moderate NO_x emission control in gas engines where a slight increase in HC could be tolerated, because of the low HC emission level typical of these engines.

5.2.1.5 Engine Speed

In general, increasing the engine speed at constant torque or constant power is accompanied by some reduction in HC. However, the degree of improvement in HC depends on several factors, including the design of the piston rings, the piston ring condition, and the blow-by flow rate.

Increasing the engine speed at constant power results in substantial reduction in NO_x. In this case engine load decreases, resulting in higher charge dilution in the engine cylinder (internal EGR). Although this technique might be applicable to stationary engines, further evaluation of the attendant affects on engine life would be required before application to stationary engines could be seriously considered.

5.2.1.6 Valve Timing

In current spark ignition engines the valve timing is set for maximum engine power output. This requires high volumetric efficiency, combined with minimum charge dilution by residual gases.

Changing the valve timing from the factory setting increases the amount of residual gases resulting in lower HC and NO_x emissions. These improvements might be achieved with little or no penalty in specific fuel consumption.

5.2.1.7 Engine Load

In most spark ignition engines NO_x decreases with decreasing engine load, while HC increases as a result of the lower average surface temperature of the cylinder walls and the attendant increase of "wall quenching." In principle, this technique is applicable to stationary engines, but it would result in a proportionate reduction in the power output of the engine, resulting in a substantial increase in engine cost per unit horsepower output.

5.2.1.8 Exhaust Backpressure

In some engines, increasing the exhaust backpressure might result in some reduction in NO_x, especially in combination with a change in valve timing. However, because of the adverse effects of increasing exhaust backpressure on engine power and fuel economy, this technique would probably not be attractive from a cost/benefit point of view.

5.2.1.9 Combustion Chamber

Combustion chamber modifications including changes in the engine head, the location of the valves and spark plug, the piston top shape, and the stroke-to-bore ratio have a direct influence on the exhaust emissions from spark ignition engines. Although the majority of these modifications could not be incorporated into existing engines, these parameters warrant further consideration for potential application to new engine designs.

5.2.1.10 Fuel System

Although considerable research and development has been devoted to the improvement of carburetors, fuel maldistribution to the individual engine cylinders remains an unsolved problem area. The electronic fuel injection systems utilized in some automobiles are claimed to improve the fuel distribution and, to some extent, the exhaust emissions and fuel economy of these engines. However, statistical data on durability, maintenance requirements, and emission control deterioration of these systems are presently lacking, and no information is available relative to the potential effectiveness of these systems in stationary engines.

5.2.1.11 Homogenation of Air-Fuel Mixture

Experimental evidence clearly shows a considerable potential for emission reduction by operating spark-ignition engines with homogenized air-fuel mixtures. In the absence of size limitations, this emission control technique appears feasible for stationary engine applications and additional development efforts should be considered in that area.

5.2.1.12 Stratification of Air-Fuel Mixture

Stratification of the air-fuel mixture, particularly the open-chamber concept, offers the possibility of substantial emission reductions as well as gains in fuel economy. However, application of this technique would require redesign of the engine and incorporation of a high pressure fuel injection system. To date this engine concept has not been evaluated for potential use in stationary engines.

Recent developments in prechamber type stratified charge engines indicate potential advantages of this engine concept in terms of improved exhaust emissions and specified fuel economy relative to conventional engines. It is conceivable that the prechamber concept might be applicable also to new and existing stationary engines.

5.2.1.13 Exhaust Gas Recirculation

Exhaust gas recirculation (EGR) is a simple and effective method of NO_x control in spark-ignition engines. Over 80-percent reduction of NO_x might be achieved by recycling 15 to 20 percent of the exhaust gas. However, this technique results in a considerable fuel economy loss and sometimes an increase in HC. If only five to ten percent of the exhaust is recycled, the fuel economy loss may be compensated for by advancing the spark timing or leaning the mixture. In this case, NO_x might be reduced by as much as 40 percent. However, a number of potential problem areas related to the long term effect of EGR on engine life would have to be resolved before incorporation into stationary engines could be seriously considered.

5.2.1.14 Water Injection

Water injection is a very effective NO_x abatement technique for spark ignition engines. If water is ingested by the engine in finely atomized form, cooling of the intake charge would improve the volumetric efficiency of the engine and would result in lower NO_x emissions. For example, an 80-percent reduction might be achieved by injecting water at a rate equal to the fuel flow. This improvement in NO_x might be accompanied by small increases in HC and specific fuel consumption.

5.2.1.15 Fuel Modifications

Conversion from liquid fuels such as the gasoline to gaseous fuels (LPG, natural gas, etc.) is relatively simple and results in a substantial reduction in the exhaust emissions. This approach appears to be an attractive one for use in stationary engines.

5.2.1.16 Thermal Reactors

Thermal reactors have been shown to be quite effective in reducing the HC and CO emissions from automotive spark-ignition engines, particularly in conjunction with other emission control techniques such as EGR and spark retard. Although NO_x is not altered in thermal reactors, these devices might be useful for those stationary engines that have high HC and CO emissions.

5.2.1.17 Catalytic Converters

Like thermal reactors, catalytic converters, when fresh, are very effective in reducing the HC and CO emissions from gasoline and gas-fueled spark-ignition engines. However, a number of potential problem areas related to catalyst performance degradation would have to be resolved before these devices could be seriously considered for use in stationary engines. At this time, there are no reducing catalysts available for NO_x control, that would be effective in the oxidizing atmosphere typical of many stationary spark-ignition engines.

5.2.1.18 Control of Emission From Blow-by, Carburetor and Fuel Tank

Positive crankcase ventilation and vapor recovery systems have become a part of the emission control systems used in current model year automobiles. These systems, which are designed to eliminate HC emissions from these sources have little impact on the operation of the engine and are directly applicable to stationary engines.

5.2.1.19 Combined Emission Controls

Several combinations of the previously discussed emission control devices are applicable to stationary spark-ignition engines. The system selection depends on many factors including engine application, engine size, type of fuel used, duty cycle, environmental factors, desired emission reduction, and permissable loss in specific fuel consumption.

Determination of the optimum emission control system for the various spark ignition engine classes requires a comprehensive optimization study considering many factors such as fuel consumption, maintenance requirements and effects on other engine components.

5.2.2 Economic Considerations

5.2.2.1 Emission Control Equipment Cost

A number of studies have been conducted to establish the economics of emission control equipment installed either as retrofit systems in existing automobiles or as accessories in new engines (Refs. 5-1 through 5-4). Although these devices were designed for use in automotive spark ignition gasoline engines, the cost figures as well as the cost-effectiveness index can serve as guideline values in estimating the installation and operational costs of similar equipment on stationary gasoline engines.

Table 5-1 (Ref. 5-5) presents the estimated sticker prices for emission control hardware for automobiles ranging from positive crankcase ventilation systems to dual-catalyst systems. Table 5-1 shows the cost of individual components and of total systems installed in particular model year vehciles. The first column presents either value added or hardware cost including material, labor, overhead, and G&A. These cost values provide a base for estimating the installation cost of similar devices on stationary gasoline engines. Some of the listed emission control components are available from the shelf; others may require modification for use in stationary engines. Of course, the cost of the modified devices might be substantially higher than for the shelf units listed in Table 5-1.

A meaningful assessment of the cost (per unit horsepower output) of emission control systems or devices for large stationary gas engines is not possible at this time. However, it is conceivable that the cost of these systems might be many times higher than for equivalent systems used in automotive gasoline engines.

TABLE 5-1. ESTIMATES OF STICKER PRICES FOR EMISSIONS HARDWARE FROM 1966 UNCONTROLLED VEHCILES TO 1976 DUAL-CATALYST SYSTEMS (Ref. 5-5)

Model Year	Configuration	Typical Hardware			
		Value Added	List Price	Excise Tax	Sticker Price
1966	PCV-Crank Case	1.90	2.85	0.15	3.00
1968	Fuel Evaporation System	9.07	14.25	0.75	15.00
1970	Carburetor Air/Fuel Ratio	0.61	0.95	0.05	1.00
	Compression Ratio	1.24	1.90	0.10	2.00
	Ignition Timing	0.61	0.95	0.05	1.00
	Transmission Control System	2.49	3.80	0.20	4.00
	Total 1970				8.00
1971-1972	Anti-Dieseling Solenoid	3.07	4.75	0.25	5.00
	Thermo Air Valve	2.49	3.80	0.20	4.00
	Choke Heat By-Pass	2.74	4.18	0.22	4.40
	Assembly Line Tests, Calif (1/10 vol)	0.18	0.57	0.03	0.60
	Total 1971-72				14.00
1973	OSAC (Spark Advance Control)	0.48	0.95	0.05	1.00
	Transmission Changes (some models)	0.63	0.95	0.05	1.00
	Induction Hardened Valve Seats (4 and 6 cyl)	0.72	1.90	0.10	2.00
	EGR (11 – 14%) Exhaust Recirculation	5.48	9.50	0.50	10.00
	Air Pump – Air Injection System	27.16	43.32	2.28	45.60
	Quality Audit, Assembly Line (1/10 vol)	0.23	0.38	0.02	0.40
	Total 1973				60.00

TABLE 5-1 (continued)

Model Year	Configuration	Typical Hardware			
		Value Added	List Price	Excise Tax	Sticker Price
1974	Induction Hardened Valve Seat V-8	0.72	1.90	0.10	2.00
	Some Proportional EGR (1/10 vol at \$52)	3.21	4.94	0.26	5.20
	Precision Cams, Bores, and Pistons	2.44	3.80	0.20	4.00
	Pretest Engines – Emissions	1.80	2.85	0.15	3.00
	Calif. Catalytic Converter System (1/10 vol at \$64)	4.02	6.08	0.32	6.40
	Total 1974				20.60
1975	Proportional EGR (acceleration-deceleration)	20.07	30.02	1.58	31.60
	New Design Carburetor with Altitude Compensation	7.52	14.25	0.75	15.00
	Hot Spot Intake Manifold	2.87	4.75	0.25	5.00
	Electric Choke (element)	2.67	4.75	0.25	5.00
	Electronic Distributor (pointless)	4.35	9.50	0.50	10.00
	New Timing Control	1.40	2.85	0.15	3.00
	Catalytic – Oxidizing-Converter	18.86	34.20	1.80	36.00
	Pellet Charge (6 lb at \$2/lb)	12.00	20.52	1.08	21.60
	Cooling System Changes	1.17	1.90	0.10	2.00
	Underhood Temperature Materials	0.63	0.95	0.05	1.00
	Body Revisions Welding Presses	0.67	1.90	0.10	2.00
	Assembly Line Changes	0.13	0.95	0.05	1.00
	End of Line Test Go/No-Go	1.85	2.85	0.15	3.00
	Quality Emission Test	1.22	1.90	0.10	2.00
	Total 1975				138.20
1976	2 NO_x Catalytic Converters[a]	22.00	37.05	1.95	39.00
	Electronic Control[a]	28.00	47.50	2.50	50.00
	Sensors[a]	3.00	5.70	0.30	6.00
	Total 1976				134.00

[a] 1976 most common configuration

5.2.2.2 Operating Cost

The general correlation trend of fuel penalty (increase in SFC) as a function of the reduction of NO_x emission for several automotive emission control systems has been presented in Figure 4-26. Although the data on which this correlation is based are results from extensive testing and evaluation of the emission control systems by several companies, an interpretation of this correlation for estimates of operating costs of stationary gasoline engines equipped with similar systems is subject to the following considerations. The emission test data were obtained on the basis of a typical car driving schedule or standard CVS test cycle. Therefore, the relative emission reduction at rated engine condition may be different. To satisfy the requirement of good car driveability, the automotive emission control systems are in most cases designed for rich air-fuel mixture operation. In stationary engines operating mostly at constant speed, leaner mixtures can be tolerated and therefore the fuel penalty associated with the incorporation of emission control systems might be lower than in the case of automotive engines. At this time, there is insufficient data available to permit a meaningful evaluation of these factors.

5.3 GAS TURBINES

5.3.1 Economics of Emission Control

Based on the discussions presented in Section 4.3, it is apparent that incorporation of emission control devices into stationary gas turbines would involve some additional costs either in the form of special R&D efforts aimed at emission reduction or for special hardware serving that purpose, or both. These additional costs, which include investment and operating costs, are expected to vary as a function of the design of the gas turbine, the locally established emission regulations, and the available resources such as water, for instance.

The cost data shown below should be considered to be average values rather than representative of a specific design.

In order to obtain reference levels for the emission control costs, investment and operating costs of stationary gas turbines operating without emission controls (except when indicated) are presented in the following subsection.

5.3.1.1 Investment Cost (Base Power Unit)

The investment cost (installation and equipment cost) of gas turbines utilized in electric power generating plants varies as a function of plant size, emission and noise levels, and location. For simple cycle peak load shaver installation in the continental United States, typical 1973 cost data are as follows: Output - $88.50/kw for 20 MW units (Ref. 5-6); $101.30/kw for 26 MW units using water injection for NO_x abatement (Ref. 5-7). The investment cost of combined cycle units for intermediate loads (500-560 MW output) varies between $132 and $168/kw (Ref. 5-8). The latter value is for a Puerto-Rico installation and comparison with other data shows that the continental United States installations would cost approximately $30/kw less. This is in agreement with Ref. 5-9 which quotes an average price for combined cycle units of $125/kw. Ref. 5-10 quotes a combined cycle cost of $130-$160/kw and recent reports for a 700 MW combined cycle plant indicate a cost of approximately $116/kw. According to Ref. 5-10, the cost of a regenerative gas turbine is higher than that of a simple cycle configuration.

5.3.1.2 Operational Cost (Base Power Unit)

The cost per kwhr energy produced is strongly affected by (assuming fixed rate of interest, depreciation, taxes, and insurance) the fuel cost which contributes approximately 50 percent for the peaking unit and approximately 80 percent for the intermediate load, and on the annual usage rate. Typically, an 80 MW simple cycle gas turbine plant operated as a peaking unit produces energy at a cost of 20 to 24 mils/kwhr. In an intermediate load installation, the cost is about 13 to 15 mils/kwhr (fuel cost 80¢/10^6 Btu, 1972 prices).

For comparison, steam electric plants in the 1000 MW size category require investment costs of the order of $300/kw. In this case, the energy cost for continuous operation is about 8 mils/kwhr, based on a fuel cost of 35¢/10^6 Btu (Ref. 5-11). Thus, it appears that gas turbines operating at intermediate load levels would approach the operating cost of steam power plants if the gas turbine could operate with cheaper fuels or in conjunction with more efficient cycles such as the combined or regenerative cycles.

5.3.1.3 Near Term Control Cost

In the absence of Federal regulations, various emission limits have been established for gas turbines by many states, counties and cities (Ref. 5-12). The interim emission controls used in these installations consist of moderate modifications to the existing combustors, supplemented by water or steam injection, if required.

5.3.1.3.1 Combustor Modification Costs

In several instances the combustor modifications were implemented over a period of several years and smoke reduction was one of the early benefits. Combustor changes require careful evaluation with respect to their effect on the life of the hot end components, since long time intervals between overhaul and long engine life are principal design requirements for stationary gas turbines. Consequently, evolution of combustors for these turbines is a slow process, although a competitive market and the national goal of tighter emission controls as now evidenced in the automotive field, will maintain the pressure for continuous improvements. The cost associated with even moderate combustor improvements has to be amortized in the cost of units subsequently sold, and the cost increase (discounting inflation and rising labor cost) will vary from about one percent for gas turbines in the 30 MW class to over 10 percent for smaller units (3 MW or less) (Ref. 5-13).

5.3.1.3.2 Water/Steam Injection Costs

Water or steam injection are other methods that have been adopted by various manufacturers (GE, GM, TP&M, Westinghouse) to meet various NO_x emission limits. In the evaluation of these concepts, cost data provided by several gas turbine manufacturers and utilities were utilized.

As previously stated in Section 5.3.1, the baseline investment cost of uncontrolled simple cycle gas turbines is about $80 to 100/kw and the operational cost (capital cost plus maintenance plus fuel), is 20 to 24 mils/kwhr for intermediate load turbines (6000 hours per year; fuel cost at 80¢/10^6 Btu). The cost increases due to water or steam injection were computed from the above baseline data. As an example, the investment cost breakdown for a water injection system consisting of combustor/injector modifications, water pump, valving, piping, storage, and water treatment equipment, as provided by San Diego Gas and Electric is shown in Table 5-2.

The investment cost of water injection can vary from approximately ten percent of the baseline cost for a 2-MW plant to approximately six percent for 49- and 81-MW plants. The investment and operating costs of steam injection systems are generally higher unless superheated steam is available from other plant sources. This is illustrated in Table 5-3, which presents average data from other sources.

As shown in Table 5-3, the water injection investment and operational cost is prohibitive for small gas turbines. However,

TABLE 5-2. WATER INJECTION INVESTMENT COST (SAN DIEGO GAS AND ELECTRIC)

Control System	Gas Turbine Size		
	20 mW	49 mW	81 mW
Combustor Modifications including Water Injection Nozzles	$1.00/kw	$0.86/kw	$1.04/kw
Water Injection Pumps and Water Regulation System	$3.54/kw	$2.88/kw	$3.10/kw
Associated Piping and Water Storage Facilities	$1.72/kw	$1.05/kw	$0.87/kw
Water Treatment Equipment	$0.90/kw	$0.47/kw	$0.47/kw
General Expenses including Engineering, Administration, Testing Taxes	$1.15/kw	$0.82/kw	$0.57/kw
TOTAL	$8.31/kw	$6.07/kw	$6.05/kw

TABLE 5-3. WATER/STEAM INJECTION COST AS A FUNCTION OF POWER PLANT SIZE

mW Output	Investment Cost, Percent Baseline		*Operational Cost, Percent Baseline	
	Water	Steam	Water	Steam
0.26 (350 hp)	100.0	150.0	55.0	165
2.90 (3900 hp)	18.0	24.0	6.5	32
20.00	10.0	12.0	6.0	–
33.00	7.3	10.6	5.7	–
65.00	7.3	10.6	5.7	–

*For peaking gas turbine, 1000 hour/year since the NO_x emissions are inherently lower for smaller gas turbine units (because of the reduced residence time of the combustion products in the chamber), water injection may not be required for this engine category, particularly if less stringent emission standards would be imposed on the smaller engines.

The operational cost of water injection decreases with increasing gas turbine usage and for intermediate loads (6000 hour/ year) it might approximate 2.5 percent of the basic operating cost of the plant.

No operational steam injection costs are available for the larger power plants. However, it appears that the higher gas turbine efficiency achieved with steam injection combined with the higher power output capability of the engine would compensate for the cost of steam generation.

The cost of water injection and its effect on NO_x abatement cost is further illustrated in the following example. In this case, a simple cycle 30 MW plant is considered having uncontrolled emissions of 220 ppm of NO_x (Figure 3-31), which corresponds to approximately 420 pound/hour of NO_2. As indicated in Figure 4-80, the use of water injection at a rate of about 80 percent of the fuel flow rate would reduce the NO_x by 75 percent, or 315 pound/hour.

For a peak shaving turbine (1000 hours/year), the operating cost would be 5.7 percent of, say 20 mils/kwhr, or $34/hour. Thus, one pound reduction of NO_x is achieved at a cost of approximately 11 cents, which corresponds to $216 for every ton of NO_x removed.

For the same gas turbine operating at intermediate loads (6000 hours/year), the water injection cost would be only $10.50 per hour or approximately 33 mils per pound NO_x reduction, which corresponds to an added operational expense of $67 per ton of NO_x removed. Thus, NO_x control becomes more cost effective with increasing load factors.

Water and steam injection is currently being used by some utilities (for instance, San Diego Gas and Electric). To date, deleterious effects have been observed on the combustors, turbine buckets, and nozzles. However, several years of operation are necessary to obtain meaningful, long-life data. Some gas turbine manufacturers feel that about two years of research work costing several millions of dollars would be required for the development of a proven water injection system.

5.3.1.4 Far Term Control Costs

As discussed in Section 4.3.3.3, the development of advanced low emission combustors and improved fuel delivery systems is required to meet future emission goals with "dry operation." The time and cost of such a development program will be substantial. It is estimated that the basic development would require approximately three to six years with an additional two to four years required for field testing. According to Ref. 5-13, over four million dollars would be required for the development of a small 3 mW gas turbine combustor. The development of a low emission combustor for a large aircraft gas turbine is of the order of $100 million (Ref. 5-14). Stationary gas turbine combustor developments will probably be close to the latter figure.

The progress reportedly achieved with externally mounted combustors (Section 4.3.3.3) cannot be overlooked. If indeed such combustors can meet the emission limits of San Diego County (see Table 3-20), an advanced, low-emission combustor may appear on the scene sooner than expected, since the highly competitive gas turbine market will not allow other manufacturers to lag far behind.

5.3.1.5 Emission Controls Evaluation

The discussion and data presented in Sections 4.3 and 5.3 permit a general evaluation of the various emission control devices for near and far term application.

The interim devices include minor combustor modification and water/steam injection. The far term controls include extensive changes in the in-line combustor, closed cycle gas turbines (see Section 3.3.1.4), and externally-mounted combustor gas turbines.

5.3.1.5.1 Interim Emission Control Devices

The modest combustor modifications consisting of leaning the primary zone, internal gas recirculation, and air-assisted fuel injection are, in many cases, already in operation. These devices are generally not sufficient to meet the more stringent emission limits, except smoke (see Table 3-20), but they are undoubtedly steps in the right direction. The incremental cost of such changes is small and the main attention must be paid to the long-term effect on the life of the hot end components.

Water or steam injection is very effective in the reduction of NO_x. It appears possible to obtain this reduction without any increase in HC or CO by tailoring the water or steam injection system to the particular combustor design. The investment cost increment for water injection equipment is acceptable, at least for units above 20 MW output, amounting to approximately ten percent of the basic unit cost. The operational cost of water injection adds only a few percent to the basic operating cost.

The effect of water or steam injection on long term engine life is not yet known, but operation up to date did not encounter any major problems.

One of the disadvantages of water injection is that it limits the siting flexibility of simple cycle and regenerative gas turbines which normally require no water supply. This is of particular importance for areas with limited water sources or subfreezing ambient temperatures. Another disadvantage is related to an attendant reduction in engine efficiency which gains importance with the rising cost of fuels.

Therefore, water or steam injection should be considered to be useful stop-gap arrangements which permit compliance with most of the existing or planned regulations, until a "dry" advanced combustor becomes available.

5.3.1.5.2 Far Term Emission Control Devices

The advanced in-line combustor with prevaporized, premixed fuel-air charge, short residence time, and lean primary combustor, when developed, will be able to meet very stringent emission regulations. However, such a combustor might not become available until the 1980s.

The closed Brayton cycle gas turbine, if successful, will probably be limited to smaller units (less than 20 MW) because of higher investment cost and greater complexity. The main advantage of this engine type is its flexibility in burning a variety of fuels, and the applicability of low emission combustion system technology developed for both steam boilers and gas turbines.

Similar comments apply to external combustors projected for use in open cycle gas turbines. In this design, the volumetric constraints of in-line combustors characteristic of mobile gas turbines are alleviated, thus permitting more effective utilization of more conventional emission control techniques, including staged combustion and flue gas recirculation. Although gas turbines with external combustors are now in production, no design and performance details are currently available from these engines. These configurations which are projected to be fully developed by the late 1970s may provide an early answer to the low emission "dry" combustor.

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SECTION 6

RECOMMENDED PROGRAMS

Based on an evaluation of the available test data, a number of emission control approaches have been identified which might offer substantial improvements in the emissions of stationary engines without harmful side effects relative to specific fuel consumption and engine life. However, significant data gaps exist in most cases and these would have to be resolved before a final assessment of the applicability of these techniques to stationary engines would be possible. The research programs outlined below are aimed at providing part of the required information.

6.1 DIESEL ENGINES

Because of the long service life of most stationary diesels, it appears that retrofitting of existing engines might be required to achieve any near-term impact on the emissions from these engines. Two potentially attractive techniques, water injection and EGR, have been identified which might be applicable to both new and existing engines.

1. Although exploratory testing of emulsified fuels has not been successful in one engine, it is suggested that this technique be further evaluated because of certain advantages in terms of design simplicity, effectiveness, and reliability inherent in this technique relative to water injection into the intake manifold. The first phase of the proposed experimental program would be conducted on a single-cylinder prechamber or open-chamber engine and would be concerned with a parametric evaluation of NO_x, SFC, and water-to-fuel flow ratio in the emulsion. In these tests, different fuels containing varying amounts of fuel-bound nitrogen would be utilized to determine the NO conversion factor under various engine operating conditions. Upon successful conclusion of this program phase, durability testing would be performed using emulsified fuels to establish the long-term effects of the water on important engine components. Another objective of this particular program phase would be the determination of the required water purity, which has a significant effect on emission control economics. If these single cylinder tests produce satisfactory results, follow-on work with multicylinder engines should be undertaken.

2. EGR has been shown to be a rather cost-effective NO_x abatement technique for diesel engines, particularly in conjunction with intake air cooling. However, there is a lack of information relative to the long-term effects of EGR on the engine and the required cleanliness of the recycle gas. It is recommended that these parameters be evaluated experimentally on a single-cylinder engine. Based on these data, an optimum EGR system configuration would be identified with respect to EGR temperature, filter system, and fuel quality requirements. Upon successful completion of this initial program phase, testing of the optimum system would be performed in a suitable multicylinder engine. Results of this program could be also extrapolated to the case of the stationary spark-ignition engine.

6.2 SPARK IGNITION ENGINES

1. The specific mass emission of NO_x from large stationary gas engines is substantially higher than from stationary gasoline engines, in spite of the fact that gas engines generally operate with very lean fuel-air mixtures. Lean operation in automotive engines normally results in reduced NO_x formation. Several causes such as high

local temperatures in the large combustion chambers, combined with relatively long residence times of the reacting species in the high temperature environment and stratification of the mixture in the combustion chamber, might contribute to the high NO_x emission from these engines. To resolve these potential problem areas, it is recommended that an experimental program be conducted on a typical low-speed gas-fueled stationary engine to determine the effects of all engine variables on NO_x, including mixture homogenation, residual gas content, wall temperature, and species residence time. These results would enable determination of changes in design and/or operation which would be required to reduce NO_x.

6.3 GAS TURBINES

1. The interim emission control techniques currently used in some stationary gas turbines require more basic research and development. In particular, the limited data available on water and steam injection show inconsistencies relative to HC and CO. It is recommended, therefore, that an experimental program utilizing a small gas turbine be conducted to determine the effect of water atomization, degree of fuel-water (steam) mixing, and water-fuel ratio on the emissions produced. Other operational parameters of the gas turbine would be monitored during the program to obtain the comparative (dry and wet) performance data required for a comprehensive assessment of water and steam injection.
2. Because of the extremely low NO_x emission potential of prevaporized, premixed combustors, (as evidenced by analysis and limited experimental modeling) and the lack of applicable test data, it is recommended that a research program be performed which is aimed at the development of such a combustor. The initial development work should be performed in the 100 to 200 hp equivalent category. The principal objectives of the proposed program would include the demonstration of concept feasibility, emission performance, combustion efficiency, and operational safety.
3. A third desirable program, which would be both analytical and experimental in nature, would be directed to a study of an externally-mounted, low-emission combustor. The goal in this program would be the demonstration of the benefits realized by the utilization of the

greater volume and design freedom associated with the external combustor. The design goals of the combustor proper are concomitant emission reduction, high combustion efficiency, and uniform outlet temperature distribution.

4. A further area for basic research is in the catalytic combustor. A program in this area is required to resolve known potential problems before their application to stationary gas turbines can be further seriously considered. This research program should explore the effects and characteristics in the area of catalyst durability, specific heat release rate, ignition characteristics, and pressure drop. Initial evaluations in small scale would be adequate. Larger scale experiments would not be recommended until a satisfactory demonstration of potential in the small scale experiments.

6.4 EMISSION INVENTORY DATA

The relative importance of reducing stationary engine emissions, and hence the relative importance of implementing control techniques for stationary engines, is largely dependent upon the total emission inventory in the region of interest. An accurate tabulation of the contribution of the various stationary engine classes is not available today. In order to evaluate the relative need to implement research programs in the stationary engine emission control area, it will be necessary to first compile a more accurate inventory of stationary engine emissions and evaluate their impact relative to other sources in the air quality control region. It is recommended that a study to develop this inventory comparison be implemented to enable more comprehensive planning for stationary engine emission control research.

APPENDIX A

The following summary table lists the total estimated installed horsepower of all stationary engines in the United States in 1971. The data listed in this table should not be used to estimate total emission contributions or fuel usage, since these factors do not reflect actual engine duty cycles.

TABLE A-1. ESTIMATED INSTALLED STATIONARY ENGINE HORSEPOWER - 1971

Application	Diesel Engine hp		Spark Ignition Engine hp		Gas Turbine hp
	Diesel Fuel	Dual Fuel	Gas	Gasoline	
Electric Power Generation	1,570,000[a]	3,710,000[a]	90,000[a]	800,000,000[b] For all Applications	29,000,000
Oil and Gas Pipelines	830,000	390,000	10,990,000		8,746,000[c]
Natural Gas Processing	–	–	2,410,000		
Oil and Gas Exploration	1,500,000	–	500,000		–
Crude Oil Production	–	–	852,000		–
Natural Gas Production	–	–	3,237,000		–
Agricultural	7,500,000	–	–		–
Industrial Process	–	–	230,000		–
Municipal Water and Sewage	465,000	–	465,000		–
TOTAL	11,865,000	4,100,000	18,774,000	800,000,000[b]	37,746,000

[a] Estimated 1970 data

[b] See Section 3.2.2.1 for basis of estimates

[c] Installed in the 1958-1970 time period

APPENDIX B

VISITS AND CONTACTS

During the data gathering phase of the study the following organizations were visited or contacted by telephone.

Organization	Primary Contact(s)
Allis Chalmers Corp. Harvey, Illinois	Mr. E.G. Goodpaster
Avco Lycoming Stratford, Connecticut	Dr. M. Bentele Mr. H. Grady Mr. G. Opdyke Mr. N. Marchionna
Caterpillar Tractor Company Peoria, Illinois	Mr. R.E. Bosecker Mr. K.J. Fleck Mr. R.D. Henderson
Chrysler Corporation Detroit, Michigan	Mr. L.K. Haddock Mr. C.M. Heinen Mr. J. Hurst
Colt Industries Beloit, Wisconsin	Mr. W.A. Brill Mr. C.L. Newton

Organization	Primary Contact(s)
Cooper Bessemer Corporation Mount Vernon, Ohio	Mr. J.W. Holmes Mr. K. W. Stevenson
Cummins Engine Corporation Columbus, Indiana	Mr. C.T.J. Ahlers Mr. R.C. Bascom Mr. A.W. Carey, Jr. Mr. P.R. Kahlenbeck Dr. W.T. Lyn
Curtiss Wright Corp. Woodridge, New Jersey	Mr. S. Lombardo Mr. F. Spindler
Diesel Engine Manufacturers Association Beloit, Wisconsin	Mr. C.L. Newton
Engine Manufactures Association Chicago, Illinois	Mr. T.C. Young
General Electric Schenectady, New York	Mr. N.R. Dibelius Dr. H.L. Hamilton Mr. M. Jarvis Mr. R.H. Johnson Dr. E.W. Zeltmann
General Motors Warren, Michigan	Mr. G.P. Hanley
Ingersoll Rand Corp. Painted Post, New York	Dr. C.K. Powell Mr. W. Hutchins
International Harvestor Company Melrose Park, Illinois	Mr. J.C. Basiletti Mr. N.G. Beck
Murphy Diesel Corp. Milwaukee, Wisconsin	Mr. L.D. Evans
Perkins Engine Company Farmington, Michigan	Mr. N. Hartwell
San Diego Gas & Electric Company San Diego, California	Mr. H. Arfman Mr. R. Miller

Organization	Primary Contact(s)
Solar, Division of International Harvester San Diego, California	Mr. R. Kress Mr. P.A. Pitt
Turbopower and Marine Systems, Subsidiary of United Aircraft Corp. Farmington, Connecticut	Mr. W. Brazel Mr. F. Cartona Mr. J. Sleeper Mr. G. Stebbins
Waukesha Motor Corp. Waukesha, Wisconsin	Mr. N. Cox Mr. M. Groenewold Mr. W. Snyder
Westinghouse Electric Corporation Lester, Pennsylvania	Mr. M. Ambrose Mr. J. Farrow Mr. R.E. Strong
White Motor Corporation Springfield, Ohio	Mr. S. Lamb Mr. J.M. Buckley
Worthington - CEI, Inc. Buffalo, New York	Mr. L. Atwood Mr. E.L. Case

APPENDIX C

UNITS OF MEASURE – CONVERSIONS

Environmental Protection Agency policy is to express all measurements in Agency documents in metric units. With a few exceptions, this report uses British units. For conversion to the metric system, use the following conversions:

To convert from	to	Multiply by
$^\circ$F	$^\circ$C	5/9 ($^\circ$F-32)
ft	meters	0.304
ft^2	$meters^2$	0.0929
ft^3	$meters^3$	0.0283
in	cm	2.54
in^2	cm^2	6.45
BTU	kcal	0.252
BTU/lb	cal/g	0.556
kW/m^3	kW/m^3	35.3
hp	kW	0.746
$lb/10^6$ BTU	$g/10^6$ cal	1.80
lb/in^2	mm Hg	51.71
lb/hr	g/hr	453.6

GLOSSARY

AAPS	Advanced Automotive Power Systems
A/F	air-fuel ratio
ANSI	American National Standards Institute
ATC	after top center
Bhp	Brake horsepower
BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
BTDC	before top dead center
CID	cubic inch displacement
CO	carbon monoxide
CRC	Coordinating Research Council
CVS-CH	constant volume sampling-cold, hot
DDA	Detroit Diesel Allison
DEMA	Diesel Engine Manufacturers Association
EGR	exhaust gas recirculation

EPA	Environmental Protection Agency
ERC	external recirculation combustor
EVC	externally vaporizing combustor
FID	flame ionization detector
GE	General Electric Company
GM	General Motors Corporation
HC	hydrocarbons
ICI	Imperial Chemical Industries
JPL	Jet Propulsion Laboratory
LAAPCD	Los Angeles Air Pollution Control District
LPG	liquified petroleum gas
LTR	Lean thermal reactor
MIT	Massachusetts Institute of Technology
MW	megawatts
NASA	National Aeronautics and Space Administration
NDIR	nondispersive infrared
NEMA	National Electrical Manufacturers Association
NO	nitric oxide
NO_x	oxides of nitrogen
PDS	phenoldisulfonic acid
ppm	parts per million
Q/I	quality/intensity
S/V	surface-to-volume ratio
R&D	research and development

RTR	rich thermal reactor
scf	standard cubic foot
SFC	specific fuel consumption
SO_x	oxides of sulfur
TDC	top dead center
TPM	Turbo Power and Marine Systems
UOP	Universal Oil Products
WOT	wide open throttle